LEHRSTUHL FÜR ELEKTRISCHE ENERGIETECHNIK

FRIEDRICH-ALEXANDER UNIVERSITÄT ERLANGEN-NÜRNBERG
TECHNISCHE FAKULTÄT

Design-guide for fluid coolers in Power Electronics

Masterarbeit

Gonzalo Pliego Prim

Erlangen
24. April 2018

Ich versichere, dass ich die vorliegende Arbeit ohne fremde Hilfe und ohne Benutzung anderer als der angegebenen Quellen angefertigt habe, und dass die Arbeit in gleicher oder ähnlicher Form noch keiner anderen Prüfungsbehörde vorgelegen hat und von dieser als Teil einer Prüfungsleistung angenommen wurde. Alle Ausführungen, die wörtlich oder sinngemäß übernommen wurden, sind als solche gekennzeichnet.

Erlangen, 24/04/2018

Abstract

This paper is orientated for the ones who are interested in solving the heat dissipation issue that occurs in the field of Power Electronics. In high voltage direct current (HVDC) transmissions the use of water as a heat transfer fluid can be inappropriate because of the strong levels of electric tension to which it is subjected. Therefore, in order to continue dissipating heat by means of a fluid (convection), other type of coolants should be used, specifically those which remain inert when subjected to high voltages. That is, dielectric fluids specially designed with good thermal properties for this particular function.

The main objectives of this document are the validation of an analytical model that can accurately predict the behaviour of the considered fluid coolants in terms of heat dissipation and pressure drop and to give the reader an overall view on how can different fluid coolants work from a thermal point of view, involving also a comparison with liquid water. The standard cooling system considered consists of a heat source that represents an average Power Electronics chip, an aluminium plate-fin heat sink and the coolant flow that removes the heat from the fins as it travels through the heat sink.

The validation of the analytical model was done by comparing it to a numerical one, specifically with simulations made with ANSYS Fluent, the CFD engineering software used world-wide for industry applications. In the first step, liquid water was successfully validated. Therefore, the analysis advanced to the second phase which brought some interesting results for the three dielectric fluid coolants analysed (3M Novec 7500, Galden HT-135 and Fluorinert FC-3283) but not so satisfactorily results for water-glycol (the other fluid studied). From that point, two parallel paths were opened: one for researching why the analytical model did not work for water-glycol as good as for the three dielectric coolants and the other one consisted in the behaviour comparison of the dielectric fluids, whose model was stated as correct.

The research on water-glycol originated the conclusion that with low Reynolds number values the analytical model had problems to predict accurately the behaviour of the cooling system. On the other hand, the analytical method was considered valid for the dielectric fluid coolants, which allowed to get a deeper view to their operation curves. The thermal efficiency curves for these three fluid coolants were plotted in diagrams as well as the curves for the volumetric flow and the pressure drop. Selecting different values for the geometric variables and for the hydraulic power permitted the obtaining of different configurations that were analysed in the diagrams.

Finally, the dielectric fluid coolants were compared between them to analyse which one of them could be more useful for a heat transfer application. They were also compared to liquid water, which obviously gave better heat transfer outputs than the dielectric ones. The thesis ends opening new paths of study after the one here defined.

Index

1 Preface

1.1 Origin of the Project

Design-guide for fluid coolers in Power Electronics is a project that was developed as a Master Thesis for the department of *Lehrstuhl für Elektrische Energietechnik* from the university *Friedrich Alexander Universität Erlangen-Nürnberg* and in collaboration with Fraunhofer´s IISB institute in Erlangen.

The idea of the topic was born after the agreement between the author of this thesis and Prof. Dr. –Ing. Martin März about a field of work that suited both interests: Fraunhofer´s and the student´s ones.

This is how thermal management in power electronics was selected as a suitable topic for developing this Master Thesis. The heating of electronic components is a real issue in many power electronic applications and the study of how to deal with it and achieve a correct heat dissipation during operation is a particularly important field of study in the industry.

The initial aim was to create a paper that could be used for consulting by each engineer who needed information about the operation data of different types of dielectric coolants for removing the heat originated by a power electronics device by means of a heat sink. Heat removing it is an everyday problem that power electronics engineers face and it is often complicated to solve as Heat Transfer is a different field of Engineering than Electronics.

1.1.1 Brief explanation of the issue that originated the thesis

The main real life application of this thesis resides in the High Voltage Direct Current (HVDC) power transmission. Therefore, it is used in such fields as renewable energy electricity transmission (an example is offshore to onshore power grid connexion), the maritime and ship industry or in long distance modern trains. It is a growing technology and a way of energy transmission that it is becoming more important year by year [Figure 1].

Inherent to the use of this technology, it exists the heat dissipation issue. During the power conversion process in multilevel inverters the electronic chips get heated in a way that could end by damaging them until failure or, in best of cases, shorten their life cycle. This is why a refrigeration circuit for cooling the components of the whole system is needed.

In this aspect, ones of the most popular and used devices are heat sinks. Usually made from aluminium or copper, they improve heat transfer by enhancing the area of transfer, which increases the heat transfer coefficient and consequently the whole heat dissipation [1].

Using water as a fluid coolant to remove the heat from the heat sink fins is one of the most used techniques, but within the context of such high voltage (there could be inverters from 10kV to 1MV, in example) and in spite of a thick insulation, water can ionize, therefore losing its capacity of cooling and producing undesirable reaction products. That is why other type of fluids shall be studied. Regarding this paper, the choice of dielectric fluids created by chemical companies (which operate in the Materials industry) that can be used at the same time as electrical insulators and help with the heat removal can be important for the target pretended to obtain.

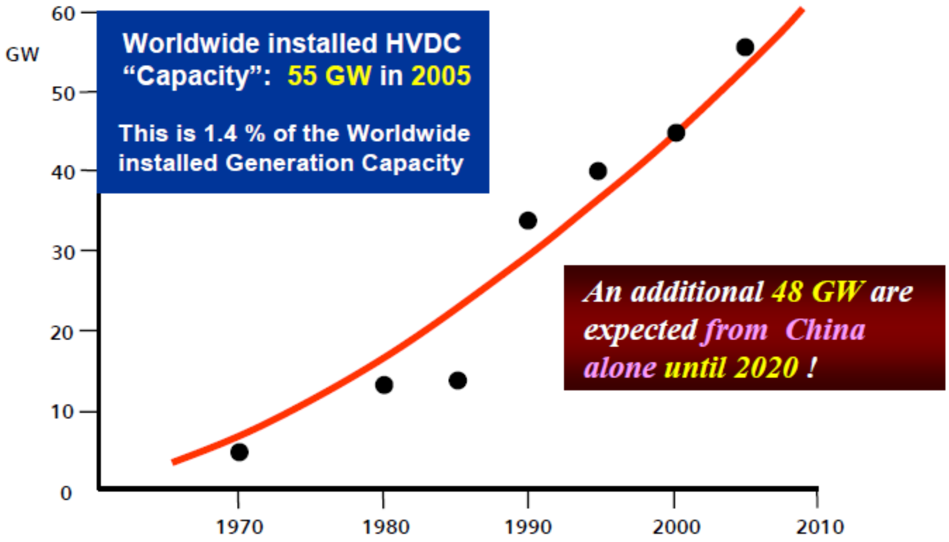


Figure 1: Worlwide installed HVDC capacity vs. years. *Source: IEEE T&D Committee 2000 – Cigre WG B4-04-2003*

This thesis has its starting point in Dr. Guan´s PhD dissertation *Thermal Analysis of Power Electronic Modules*. In the mentioned document, the author did an extended analysis of heat dissipation using aluminium heat sinks and water as a cooling fluid. It served as a touchstone for *Design-guide for fluid coolers in Power Electronics*.

1.2 Motivation

About the facts that motivated the realisation of this project, they should be divided in the two: firstly, from Fraunhofer´s side and secondly from the writer´s preferences and background studies.

On one hand, the Institute had the interest in the study of new dielectric fluid coolants in Power Electronics and the comparison of their operation to water, which it is the fluid currently used. This approach could allow observing the viability of using these new materials (which are introduced afterwards in these pages) in real life industry application.

On the other hand, as an Industrial Engineering student with a mention in Energy, for the author of this text, it was interesting to write about a subject with direct application in the energetic sector and to approach it from the heat transfer field, which is one of the main points of the mentioned specialisation. Although Electronics it is not directly in the background as the main field for the writer, it was also a challenge as it would mean a reason to learn more about the topic.

1.3 Previous requirements

In order to succeed in the realisation of the thesis, few previous requirements were to be considered.

Related to IT tools, it was important to have a minimum knowledge of CFD software for simulating fluid domains. However, a part of the process of setting the simulations was used to learn the features of the program that remained unknown at the start of the project.

In terms of the software, it was used an academic version of ANSYS CFD and Mechanical analysis, which was enough to obtain correct and accurate results of each of the different volumetric systems that were simulated.

Moreover, for the analytical approach the mathematical software MathCad was used. This tool was useful in order to solve the analytical methods in a quicker way than it would have been without it. It permitted the saving of important amounts of time by automatizing processes. As it is an intuitive tool, it was easy to understand in a matter of days most of the possibilities that it offers.

Finally, previous knowledge of heat transfer theory and its topics as convection or conduction as well as fluid mechanics, which were learned in different subjects during the Bachelor´s and Master´s Degrees were basic for being able to reach the targets of the thesis.

2 Introduction

In the next lines the objectives of the thesis are going to be set in order to have a clear idea of which were the main points that were targeted since the beginning. It is also going to be discussed the final scope of the project and the way this document is organised.

2.1 Main objectives of the thesis

In overall the main objective of this Master Thesis was to study the fluid cooling of Power Electronics components by means of a heat sink device for enhancing heat transfer and a fluid coolant for removing the heat.

In order to reach this target a route of work was determined. The study of the heat dissipation would be done via analytical equations, obtained from experimental correlations that different important figures in the history of Heat Transfer have been obtaining. Afterwards, these solutions would be compared to the results obtained by computer simulation tools, specifically ANSYS Fluent software. The studied system should always be subjected to the same conditions for a real comparison.

The main reason for choosing this path of work was that the final objective was to validate the analytical method with the computer simulations. Once validated, it is totally useful to have the right correlations and thereafter procure the operation data and curves for analysing in a visual and fast way the optimum geometry of the heat sink under determined conditions.

With that said, the final objective was to prove valid the analytical model (in chapter 4 explained) for new dielectric fluid coolants. This would make possible to draw adjusted operating curves of them, therefore allowing the extraction of useful conclusions on the real convenience of using these materials and if so, under which system conditions should be used and which would be the optimum for each one of them.

Finally, this is useful in order to open the path of study to real industry applications. Proving the viability of the usage of dielectric fluids in further HVDC power transmission contexts would be an important step for Fraunhofer IISB to begin to investigate the way to introduce it in such systems.

2.2 Specific objectives of the thesis

With the purpose of taking a closer look to the objectives of the project, the steps followed for reaching them are described in this subchapter.

Firstly, the milestone for successfully advancing was to validate Dr Guan´s theoretical model (explained in detail in chapter 4) with computer simulations. This model involved the cooling the electronic chip using liquid water as a fluid. The model would be considered valid if the difference between the simulation results and the analytical ones was in between a given interval (usually between (10% and 20%).

After proving this theoretical model with water, the same model should be exported and tested with the new fluids studied. As these are different materials, they have different thermal and mechanical properties. Thus, viscosity, density, conductivity and specific heat differ from one to another. That is why the main parameters that are involved in the analytical method of study change, so the new situation varies and that is the reason why the theoretical model must be tested again with the new fluids.

Once obtained the analytical results and the software-simulated ones, the next step conducted to compare them and extract the conclusions for accepting or refuting the model. Up to this point, two ways could appear in the followed path: either the hypothesis of the validity of the model was accepted and consequently the test was considered as a success, or the hypothesis had to be refused because of the divergence of the results, therefore a new investigation should be opened in order to create a new model or modify the one existing so that could be accurate to the reality.

The last step that was considered as an initial objective was the study of the behaviour of the different fluids when they are subjected to different initial and boundary conditions. The reason behind doing this, was that it could allow the observation of the way in how the variations in the working conditions affect their operating parameters. Hence, it can be obtained the optimum heat sink geometries for each fluid and also the range of temperatures or of volumetric flows that would be most suitable for each of them.

2.3 Scope of the project

A necessary scope of the project was intended to be fixed at beginning. Although in a thesis with a research nature it can be difficult to fix the limits right at the start, a series of restrictions were taken into account in order make the resolution of the project possible.

First of all and in consequence of this statement, the study of heat sinks was restricted to plate-finned-type ones [Figure 2]. This type of geometry permits different modifications that change the results of its operation. It was considered to study just the variation provoked by the fin thickness and the distance between fins.

Referred to the fluid coolants that were studied, the scope was closed into four:

- Water-glycol
- Galden HT-135
- Novec 7500
- Fluorinert F-3283

These four fluids were the analysed ones.

In terms of length of the project, for reasons of number of hours expected to be used in it, the scope was set to the obtaining of the operation curves of the fluids and the analysis and comparison of them subjected to different working conditions. The investigation of new experimental correlations of heat transfer parameters was left out of the scope.

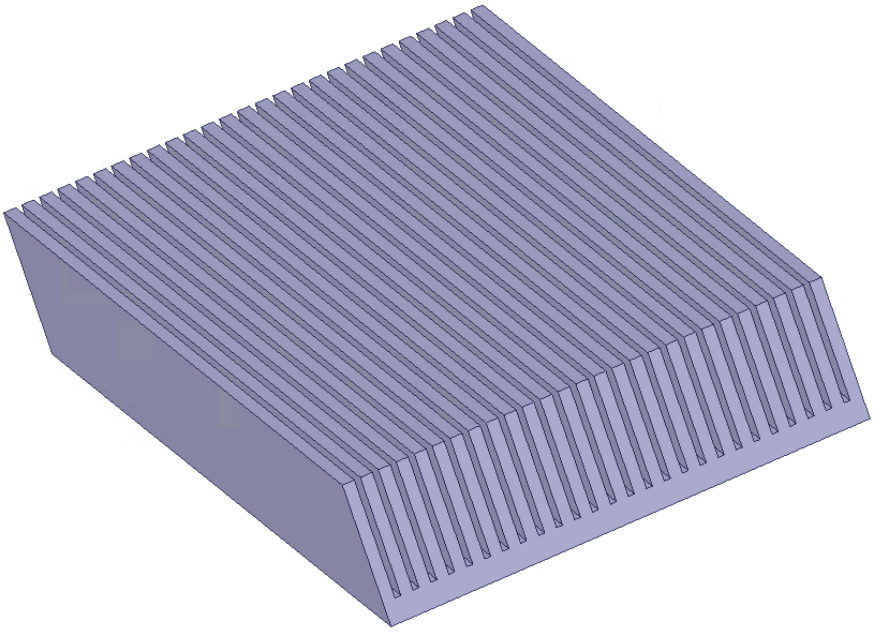


Figure 2: Image of a basic geometry plate fin heat sink. *Source: Own illustration*

2.4 Organisation of the thesis

This thesis is divided in eight chapters. Chapter 1 provides information about the origins and the motivation that led to the starting of this thesis. Moreover, it talks about the previous requirements, either knowledge or software, that were necessary to develop it.

Chapter 2 introduces the reader to the topic of the thesis with an explanation of the main objectives along with the collateral objectives that were set at the beginning. It also marks the limits of the project fixing the scope that was tried to complete.

Chapter 3 gives a brief explanation of the fluid coolants that were studied, the reason they were chosen and it also shows the thermal properties that were used and how they weree obtained.

Chapter 4 explains the theory and the steps of the analytical method of calculation. It describes the process followed and the expressions used for it. It introduces the non-dimensional parameters used and the correlations that were assigned to each one of them. In the same way, it sets the parameters that were considered constants and the ones that were considered variables and it defines the two target parameters that were the final objective of the calculations (temperature at the outlet of the fluid (T_{cf}) and pressure drop (Δp)). The boundary conditions, the simplifications of the model assumed and, finally the initial values set for the constant parameters are also described in this chapter.

Chapter 5 is analogous to Chapter 4, but it explains the numerical method of calculation. It begins with a brief explanation of the software used (ANSYS) and it continues with its configuration: the geometry created, the mesh used and defined, the set-up of the models and at last the solution and the post-processing tools applied.

Chapter 6 is the longest point of the thesis and shows the results of all the calculations together with their comparisons. It starts with the validation of liquid water analytical hypothesis and then it continues with the calculation of water-glycol as a fluid coolant and with the three dielectric fluids. The results are shown numerically and also in a visual way for the reader to achieve a perfect comprehension of everything.

In chapter 7 it can be found a different number of diagrams that describe the thermal behaviour of the different fluid coolants subjected to different conditions. The user can get from them an idea of how the geometry of the heat sink and the hydraulic power applied to the cooling circuit affects the heat transfer under different restrictions and which are the best geometries in each case. It also contains the conclusions to which the author arrived to.

Chapter 8 is the end to this thesis and makes a summary of what it has been along with the main conclusions extracted. It also describes the path that opens after this thesis to continue studying about it.

3 The fluid coolants

The fluid coolants analysed were one of the most important points of the thesis. There were initially three fluid coolants chosen (apart from water) and in the end a fourth one was aggregated (water-glycol). In the next lines a brief description of each one can be found together with an explanation of the reasons of their utility for Fraunhofer´s interests.

Before introducing the coolants, it must be explained the criteria that were followed for choosing them:

- The fluid must have dielectric properties which means that it is an electrical insulator useful in high voltage application.
- It should be designed with the main purpose of being used in heat transfer applications, therefore it should have good thermal properties for heat transfer.
- The maximum operating temperature should be higher than 105°C or higher, which is the temperature that was considered for the chip. The reason of the temperature is explained in chapter 4.

From these fluids, in order to study their capacity of dissipate heat, it was interesting to obtain there following four properties: density, conductivity, viscosity and specific heat.

With that said, the following fluids were the chosen ones.

3.1 Galden® HT-135 (2)

Galden HT-135 is a heat transfer synthetic fluid created by the chemical company Solvay. It belongs to the family of Galden® PFPE Perfluoropolyether Fluorinated Fluids and its main applications are the Electrical & Electronics and Semiconductors markets. It complies with the features of having dielectric properties, a high thermal stability, chemical inertness and it can be used in a wide operating temperature range.

In particular, Galden® HT135 belongs to the subfamily of Low Boiler Grades and it has a boiling point of 135°C. The operating temperature range is [-65°C, 125°C] so it meets the previous requirements [Figure 3].

The properties required were obtained by observing its data sheet (3). In the case of the viscosity (the most influential of all the properties), a valid equation dependant on the temperature was obtained from a graphic provided by the company Solvay []

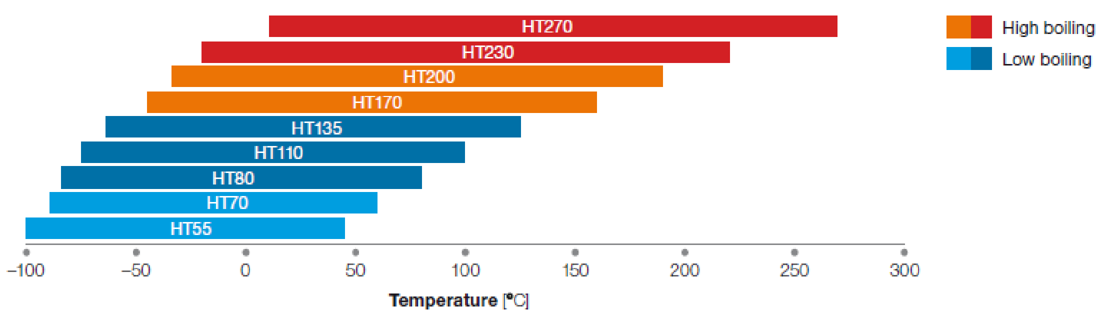


Figure 3: Operating temperature range of Galden HT fluids. *Source: Solvay (3).*

In Figure 3 it can be observed the different temperature ranges of Galden® High Boiling and Low Boiling coolants. As it has been said, HT-135 meets the operation temperature requirements which are needed to be satisfied.

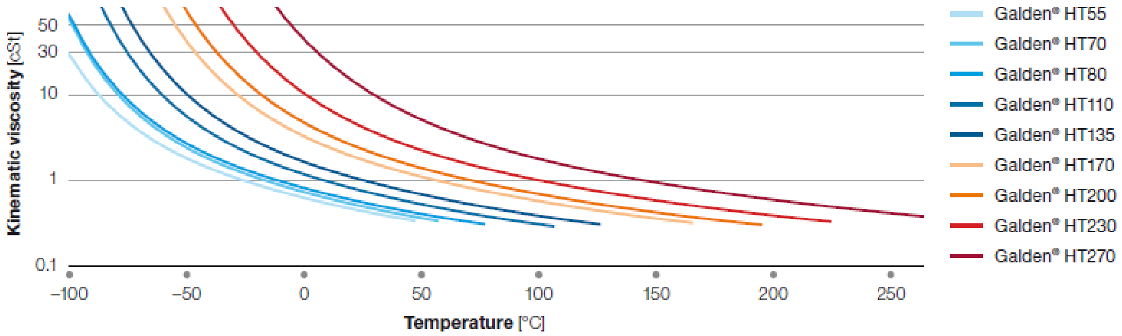


Figure 4: Kinematic viscosity vs. temperature Galden HT fluids. *Source: Solvay (3).*

Figure 4 shows the representation of the kinematic viscosity of the different Galden coolants. The equation obtained for Galden® HT135´s kinematic viscosity, using polynomial approximation of sixth grade, was the following:

$$\upsilon_{GALD}(T)=2\cdot10^{-12}\cdot T^{6}-1\cdot10^{-9}\cdot T^{5}+2\cdot10^{-7}\cdot T^{4}-2\cdot10^{-5}\cdot T^{3}+8\cdot10^{-4}\cdot T^{2}-4{,}17\cdot10^{-2}\cdot T+1{,}8$$

[Equation 1]

3.2 3M™ Novec™ 7500 (4)

Secondly, the family products of Novec™ were marked as interesting. These fluids are designed by the company 3M for diverse finalities, but within the interests regarding this thesis, it exists a group of them called Engineered Fluids that are created especially for heat transfer applications (4).

In the same way as with the previous fluid, the operating range of temperatures was observed and then Novec™ 7500 was selected [Figure 5].

The equations for the properties were also obtained from 3M webpage (5), and they are the following:

$$Z(T)=10^{\left[10^{11{,}843-5{,}0874\cdot\log[(T+273{,}15)]}\right]}$$

$$\upsilon_{NOV}(T)=[(Z(T)\text{-}0{,}7\text{-}\exp[\text{-}0{,}7487\text{-}3{,}295\cdot(Z(T)\text{-}0{,}7)+0{,}6119\cdot(Z(T)\text{-}0{,}7)^2\text{-}0{,}3193\cdot(Z(T)\text{-}0{,}7)^3]]$$

[Equations 2]

$$\lambda_{NOV}(T)=(0{,}069\text{-}1{,}798\cdot10^{-4}\cdot T+4{,}24\cdot10^{-7}\cdot T^2)$$

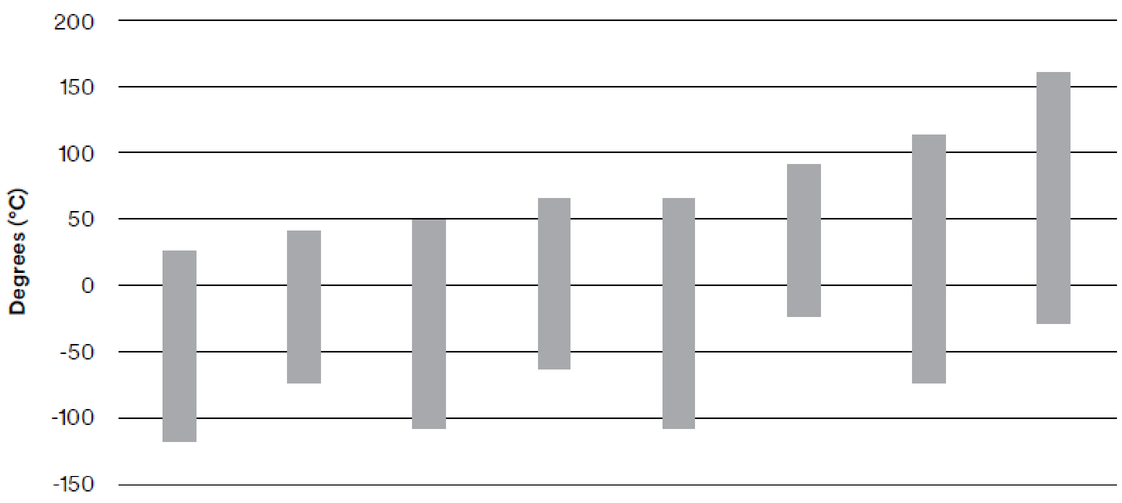
[Equation 3]

$$\rho_{NOV}(T)=(\text{-}2{,}085\cdot T+1665{,}8)$$

[Equation 4]

$$c_{NOV}(T)=(1{,}4982\cdot T+1091)$$

[Equation 5]



	Novec 7000	Novec 649	Novec 7100	Novec 774	Novec 7200	Novec 7300	Novec 7500	Novec 7700
Maximum	25	40	50	65	65	85	115	155
Minimum	-120	-75	-105	-60	-105	-30	-75	-30

Figure 5: Operating range of temperatures 3M Novec Engineered Fluids. *Source: 3M* (4).

As it can be observed in Figure 5, Novec 7500 meets the operating temperature requirements needed.

3.3 3M™ Fluorinert™ FC-3283 (6)

For ending the selection of inert dielectric fluids, another one from the company 3M was chosen. In the same direction than the others, it is a fluid used in many single phase heat transfer applications in the semiconductor manufacturing industry (6).

Its liquid range (-50°C to 128°C) makes it useful for the application in this paper discussed.

Its properties were obtained by the equations observed in its data sheet (6), and are the following:

$$\lambda_{FLUO}(T)=[0{,}065\text{-}0{,}000056\cdot(T)]$$

[Equation 6]

$$\rho_{FLUO}(T)=1878-2{,}455\cdot T$$

[Equation 7]

$$c_{FLUO}(T)=1014+1{,}544\cdot T$$

[Equation 8]

The viscosity was valued using the following curve [Figure 6], provided by the company 3M™.

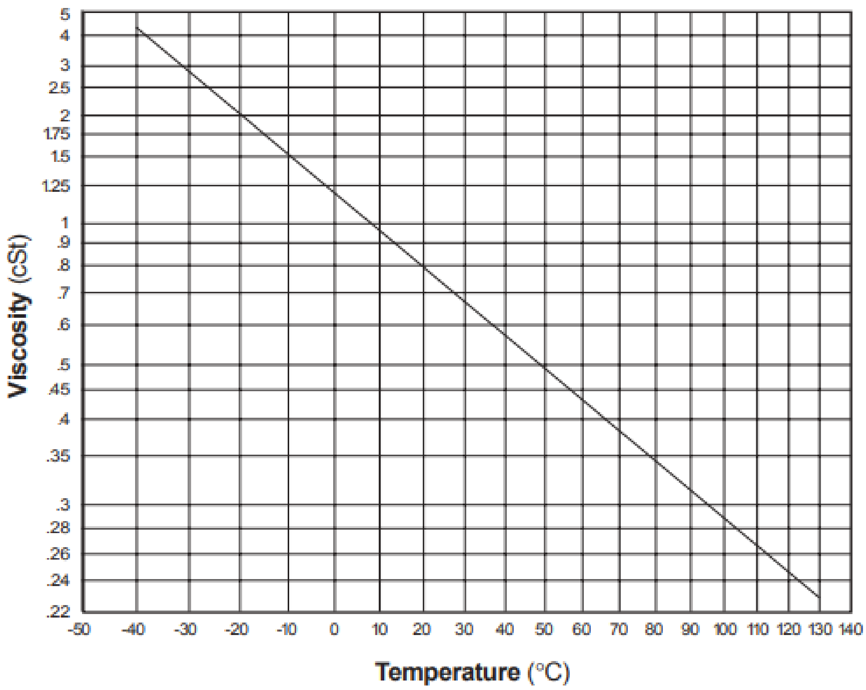


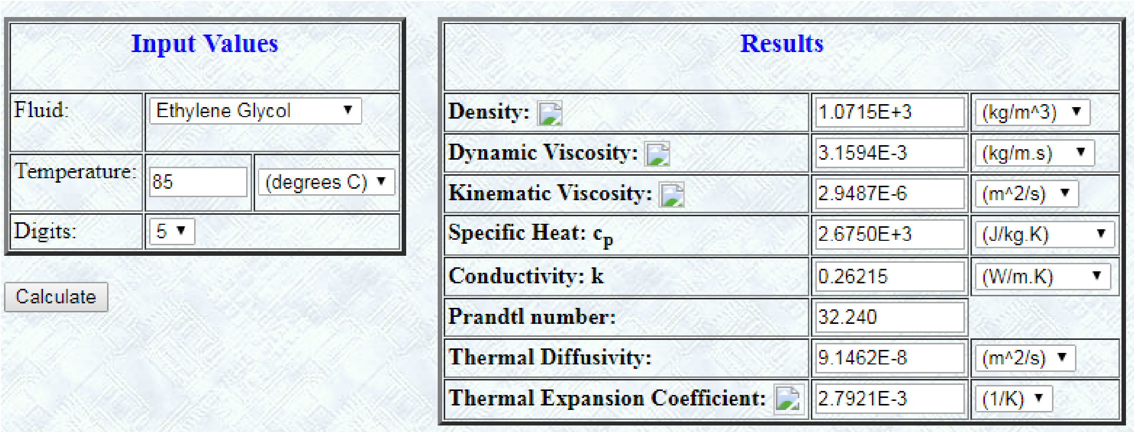
Figure 6: Viscosity vs. temperature in Fluorinert FC-3283. *Source: 3M (6)*

3.4 Water-glycol

If firstly were the previous three mentioned fluids the ones which would had been analysed, during the progress of the project it was declared profitable to study also the behaviour of water-glycol, as it is a fluid widely used for electronic cooling applications and it would be interesting for Fraunhofer IISB to also have a study of it.

Ethylene glycol is a heat transfer fluid mostly used in applications where water can freeze. Although it has lower heat transfer efficiency than water, it is useful to add it to the water solution when freezing can occur. As it will be seen in chapter 6, it has a higher density, therefore it would need a higher flowrate, thus a higher hydraulic power will be needed that would led to a higher pressure drop. One of its main applications resides in the automotive sector for an anti-freezing cooling circuit (13).

As the concentration of glycol in the solution increases, the thermal performance of the heat transfer fluid decreases. Therefore, it is better to use the lowest possible concentration of glycol necessary. For this thesis it was analysed water-glycol with a concentration of 50% of ethylene, as it was considered as a standard value for cooling applications. The properties of water-glycol were obtained directly from an internet source (14).



Input Values		
Fluid:	Ethylene Glycol	
Temperature:	85	(degrees C)
Digits:	5	

Calculate

Results		
Density:	1.0715E+3	(kg/m^3)
Dynamic Viscosity:	3.1594E-3	(kg/m.s)
Kinematic Viscosity:	2.9487E-6	(m^2/s)
Specific Heat: c_p	2.6750E+3	(J/kg.K)
Conductivity: k	0.26215	(W/m.K)
Prandtl number:	32.240	
Thermal Diffusivity:	9.1462E-8	(m^2/s)
Thermal Expansion Coefficient:	2.7921E-3	(1/K)

Figure 7: Physic properties obtained for water-glycol. *Source: www.mhtl.uwaterloo.ca. (14)*

Figure 7: Physic properties obtained for water-glycol. Source: www.***mhtl.uwaterloo.ca. (14)*** shows the print illustration of the process for obtaining the physic properties of ethylene-glycol.

4 Analytical method

In this chapter it can be found a brief explanation of the theory in which the analytical method is based. There are also highlighted all the equations and correlations that have been used, as well as the hypothesis and simplifications that have been taken into account, the restrictions and conditions applied and the process for solving each of the cases proposed.

4.1 Plate fin calculation

As it has been mentioned in 1.1, Dr Guan´s thesis *Thermal Analysis of Power Electronic Modules* is the literature from which it has it start this current paper. In it, the author makes an analytical, numerical and experimental analysis of the thermal behaviour of plate fin heat sinks using water as a fluid coolant. In the following lines, it is explained the main theory he based his calculations and the results he arrived to.

First of all, it has to be mentioned that the cooling technique involved in plate-fin heat sink devices is heat dissipation by forced convection. This means that a fluid is forced (with a fan, pump or similar) to move through the body of the heat sink in order to improve convection by increasing turbulence. The existence of the fins makes an enlargement of the area of convection, thus enhances heat transfer.

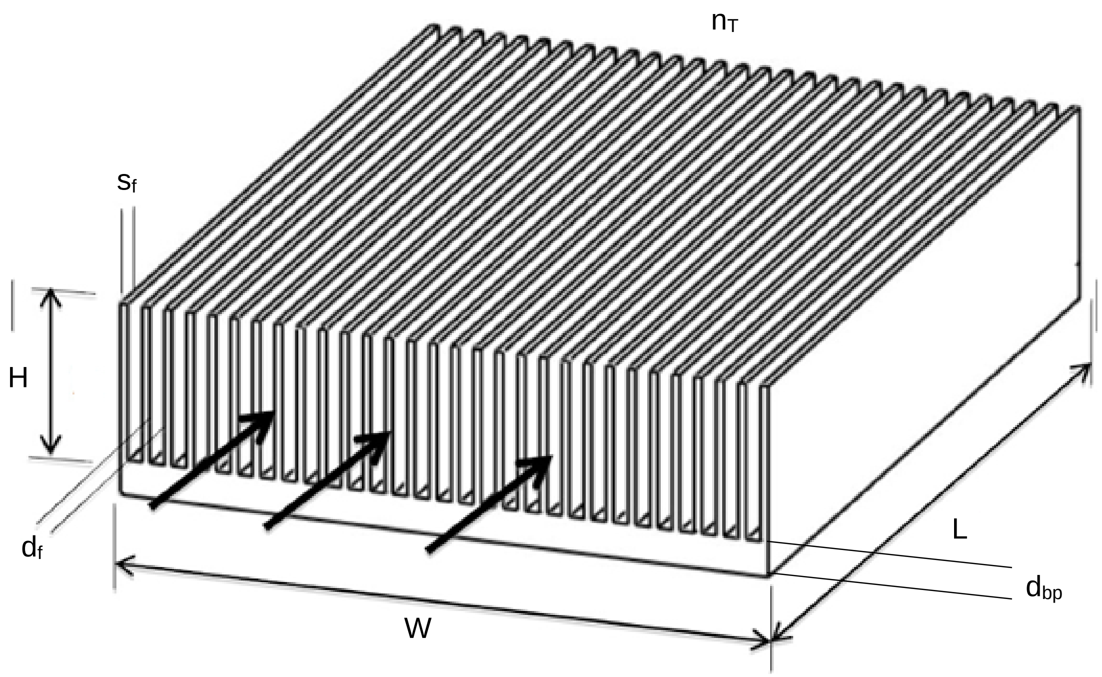


Figure 8: Heat sink schematic drawing. *Source: www.mdpi.com.*

The fins of the device, although they have the same thermal properties like the rest of the body of the heat sink, they have a loss of convection ratio with the progression of its height. This is because the temperature at its tip will always be lower than at its base due conduction process through each fin. For this reason, the efficiency (η) of the fin has to be considered as a parameter between 0 and 1 that multiplies the total fin area of convection. The following equations are directly extracted from Dr. Guan's work (1) and represent numerically what has been previously explained. Figure 8 shows the geometrical dimensions represented in a heat sink illustration.

$$\eta=\tanh\left[H\cdot\sqrt{\frac{2h}{\lambda_{fin}}}\cdot\left(\sqrt{\frac{2h}{(\lambda_{fin}.s_f}}^{-1}\right)\right]$$

[Equation 9]

$$A_t=(W-n_T\cdot s_f)\cdot L+2n_T\cdot H\cdot L\cdot\eta$$

[Equation 10]

[Equation **9** defines the calculation of the fin efficiency and [Equation **10** the total area of convection (1). Being *H* the height of the fin, *h* the convection coefficient, λ_{fin} the conduction through the fin, s_T the thickness of the fin, *W* the width of the plate fin heat sink, n_T the total number of fins and *L* the length of the heat sink.

Concerning the calculation of the thermal resistance referred to the area of convection, for a heat sink with constant heat flux applied to the backside of its base plate it can be written as [Equation 11]:

$$R_t=\frac{d_f}{\lambda_f\cdot A_b}+\frac{1}{m\cdot c_p}+\frac{1}{h_{eff}\cdot A_t}$$

[Equation 11]

Where the first term in the right side is caused by conduction effect, the second one by the heating resistance of the fluid and the last one by the convective resistance (1). Here, A_b represents the area of the base plate and h_{eff} the effective convection coefficient.

In order to leave the thermal resistance equation depending on the geometrical parameters of the heat sink and the fluid flow as well as the temperature, the equation is re-written as the following [Equation 12]:

$$R_{th}=\frac{d_{bp}}{\lambda_{fin}\cdot W\cdot L}+\frac{1}{\rho_c\cdot c_c\cdot V_{flow}}\cdot\left[1+\frac{Re\cdot Pr}{Nu}\cdot\frac{H}{L}\cdot\frac{1}{\left[1+2\cdot\sqrt{\frac{s_f}{d_f}}\cdot\sqrt{\frac{\lambda_{fin}}{\lambda_c\cdot Nu}}\cdot\tanh\left[\sqrt{\frac{H}{d_f}}\cdot\sqrt{\frac{H}{s_f}}\cdot\sqrt{\frac{\lambda_c\cdot Nu}{\lambda_{fin}}}\right]\right]}\right]$$

[Equation 12]

With Equation 12 it is possible to calculate the total thermal resistance with previously fixed geometry parameters and the known thermal properties of the fluid.

Referring to the non-dimensional parameters, the definitions used are the following:

Reynolds number: $Re_{df}=\frac{u_c \cdot D}{\upsilon}$

[Equation 13]

Nusselt number $Nu=\frac{h \cdot D}{\lambda}$

[Equation 14]

Prandtl number: $Pr=\frac{\upsilon \cdot \rho \cdot c_p}{\lambda}$

[Equation 15]

Where u_c represents the flow velocity, D is the characteristic dimension of the heat sink which in the case of plate-fin heat sinks is the distance between fins (d_f), λ the conductivity of the fluid, υ is its kinematic viscosity and ρ its density.

Then, with the last four equations it can be possible to calculate the thermal resistance once knowing the thermal properties of the fluid, the geometry of the heat sink and the volumetric flow (velocity) at the inlet of the heat sink.

The reason of leaving the calculation of the thermal resistance depending on the geometrical parameters is that it allows the investigator to know how accurately different plate fin geometries dissipate heat just by changing the geometry parameters in the equations. This facilitates the process of choice of the heat sink geometry for a particular application.

4.2 Nusselt correlation hypothesis

One of the most discussed issues in Heat Transfer has always been which Nusselt number correlation must be used in order to obtain the most accurate results possible. Having in mind that there exist a high number of correlations obtained by a large number of investigators during time, for an also large number of applications, geometries, fluid flows or types of convection, there are still applications where not even the whole historical theory can accurately predict.

In terms of plate-fin heat sinks, which are the devices concerning this thesis, either Dr. Guan in his doctorate project (1) and Pr. Dr.-Ing. März in his investigations (7), concluded after experimenting and simulating with water as a coolant that in first instance Dittus-Boelter correlation was the most accurate for plate fins and that Filonenko and Culham could predict also accurately the loss of pressure through the heat sink.

Dittus-Boelter Nusselt correlation: $Nu=0{,}023 \cdot Re^{0,8} \cdot Pr^{0,4}$ $10^4<Re<1{,}2 \cdot 10^5$

[Equation 16]

Filonenko friction coefficient correlation: $f=(1{,}58 \cdot \ln(Re_{Dh})-3{,}28)^{-2}$ $2300<Re<28000$

[Equation 17]

Culham friction coefficient correlation: $f=0{,}316 \cdot Re_{Dh}^{-1/4}$ $Re>5000$

[Equation 18]

It is one of the objectives of this thesis to validate this equations using the ANSYS software simulations, which would add new information to the two mentioned researchers' investigations.

4.3 Parameters of study

As of analysing the thermal behaviour of the different coolants, two parameters of study were targeted. Both are interesting for the concerning field of study and permit the obtaining of valid conclusions. These were the final temperature of the fluid coolant (T_{cf}), that is the one the fluid is at at the outlet of the heat sink body, and the pressure drop (Δp) of the fluid as it flows through the heat sink.

The final temperature of the fluid coolant is obtained using the following equations:

$$P_{th}=\dot{m}\cdot c_p\cdot(T_{cf}-T_{ci})$$

[Equation 19]

$$P_{th}=\frac{\Delta T_{max}}{R_{th}}$$

[Equation 20]

Where [Equation 19] is the definition of the thermal power dissipated (q) due to the increase of temperature of the fluid flow and [Equation 20] is the thermal power dissipated in terms of the difference of temperatures between the fluid flow at the inlet and the baseplate of the heat sink (ΔT_{max}) and the thermal resistance referred to the surface of convection (R_{th})(1).

The following equation defines the pressure drop:

$$\Delta p=\rho_c\cdot\frac{L}{2\cdot D_h}\cdot u_c\cdot f$$

[Equation 21]

Where D_h represents the hydraulic diameter of fluid domain as it flows through the heat sink's body.

4.4 Process of calculation

Once seen briefly the theory in which it is based the analytical method of calculation, it is time to explain the process followed for the procurement of the results. With that purpose on target, the next sub-chapters resume and describe the system domain considered, the boundary conditions set, the simplifications that were assumed for an easier solving of the problem and the steps followed during the process of calculation.

4.4.1 System domain

The system studied comprises three parts, all of them represented in Figure 9. The first of them is the heat sink, which is designed with a particular length (L) and width (W). This heat sink, which is a plate-fin design, is subdivided at the same time in two components: the base plate and the fins. The base plate connects the chip to the fins and has a known thickness (d_{bp}). Moreover, the fins grow perpendicular to the base plate and own the following parameters of interest: height (H), thickness (s_f) and distance between them (d_f).

The chip, which can be simply represented as a two-dimensional heat source, is located on the top side of the base plate. Finally, the fluid domain includes the space between fins (where the flow circulates) and its boundaries are the inlet, just in the same plane as the front face of the heat sink, the outlet, at the same plane as the back face of the heat sink, and its height, which is the same as the ones the fins have (H).

With that said, it can be noted how heat dissipation process works. The heat flows from the heat source (chip) to the plate fin base plate and from there it is conducted through the fins that increase the convection area. Finally, the fluid flow removes the heat as it travels along the space between the fins.

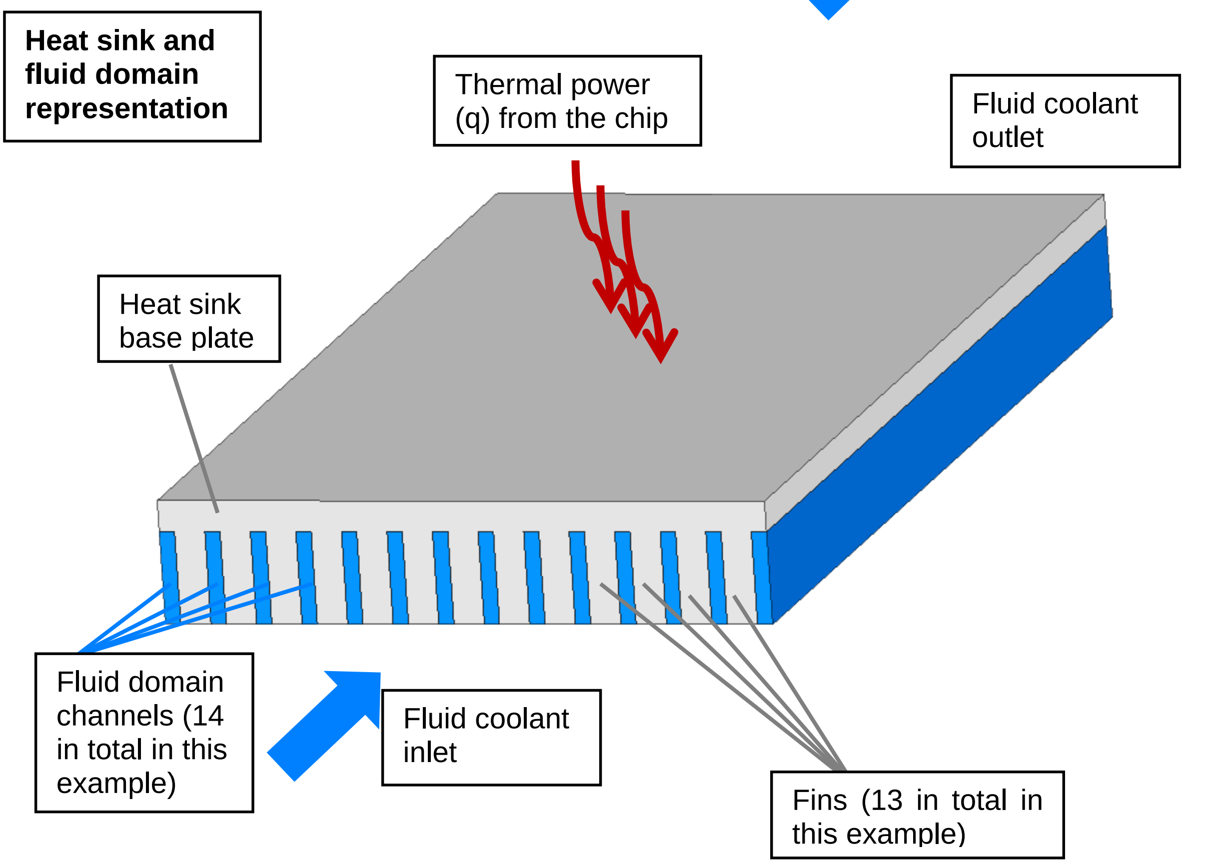


Figure 9: Representation of the whole system domain studied. *Source: Own illustration.*

4.4.2 Constants and variables

As it is explained in 4.1, the equations with which the two parameters of study were calculated, are left depending on the geometrical dimensions of the heat sink (L, W, H, d_{bp}, s_f, d_f), the temperature of the fluid flow (T_c) and the velocity of it (u_c). These parameters can be changed systematically with the aim of procuring different configurations of the system, therefore different solutions.

Due to the will of accomplishing with a fixed scope, in this thesis some of the mentioned parameters were treated as constants and the rest were left as variables so that they could be modified as of part of the concerning study. These lines are clarified in Table 1 and the explanation that follows it.

Constants	Variables
Heat sink lenght (L)	Velocity of the flow (u_c)
Heat sink width (W)	Fin thickness (s_f)
Base plate thickness (d_{bp})	Distance between fins (d_f)
Height (H)	
Temperature of the chip (T_0)	
Temperature at the inlet of the flow (T_c)	

Table 1: Constants and variables considered for the calculation

As it can be observed in the left side of Table 1, either the heat sink's length or the width are fixed as constants. This is because it was wanted to study an average measure of base plate used in Power Electronics that then could be compared in successive analysis.

Referring to the base plate thickness and the height of the fins, these parameters could be subject of variations as they have influence in the thermal behaviour of the system. Nevertheless, as a matter of accomplishing with the fixed scope, were determined as constants. The choice of these two parameters as constants was due to a practical real life situation (also explained by Dr. Guan in his thesis (1)). That is, in average Power Electronic applications the height of the heat sink is fixed because of the need of optimizing the small space available, hence is more important for real applications to procure results with fin thickness or distance between fins as variables.

Analogously, either the temperature of the chip or the temperature of the fluid flow at the inlet were fixed as constants. This decision was made based on a simplification explained in 4.4.4. It is also a crucial point if the analysis is wanted to be made around the geometry dimensions of the heat sink. The temperatures were defined so that the difference between them was of 20ºC (4.4.4).

Finally, as variables subjected to be modified there are: the fin thickness, the distance between fins and the velocity at the inlet (or the volumetric flow). Observing the case of the velocity inlet (u_c), it is a fact that it is related to the volumetric flow with the following equation:

$$u_c = \frac{V_{flow}}{H \cdot W} \cdot \left(1 + \frac{s_f}{d_f}\right)$$

[Equation 22]

Which it is also related to the hydraulic power used to impulse the fluid flow along the refrigeration cycle. That is represented numerically in the next equation:

$$P_{hyd}=\Delta p \cdot V_{flow}$$

[Equation 23]

[Equation **22]** represents the average velocity of the flow across the heat sink and [Equation **23]** shows the relation between the hydraulic power and the volumetric flow. So, in terms of practical usage, it is useful to set a P_{hyd} and to modify it between values that are accepted in average Power Electronics cooling circuits.

4.4.3 Boundary conditions

Once set the constants and variables that were supposed to rule the system, it is necessary to discuss the boundary conditions to which the system is subjected. These are the following:

- Velocity inlet (u_c): A known velocity magnitude at the inlet of the fluid domain was needed to be set in every system configuration tried.
- Temperature at the outer side of the base plate (T_0): This temperature was set in representation of the heat source (chip) (see 4.4.4).
- Temperature at the inlet of the fluid domain (T_{c0}): The temperature at which the flow is at at the inlet is also a boundary condition and it is fixed 20°C beneath T_0.
- Adiabatic boundary walls: The boundary walls that separate the system from the surroundings are considered adiabatic. That means that there is no heat transferred from the system to the outside or from the outside to the system.

4.4.4 Simplifications assumed

In order to make the analytical calculations easier and quicker without altering in a decisive way the results, some simplifications were introduced to the system. In the next lines these are explained:

- Chip´s heat source as fixed temperature (T_0): In 4.4.2 it has been mentioned that the temperature of the chip is considered as a constant. To explain the concept of *temperature of the chip* it is needed firstly to say that it is a simplification of the chip´s heat source.

  In the considered system domain, the chip is a heat source that is dissipated first through the heat sink and afterwards by the fluid coolant. But as a simplification, the heat source produced by the chip is considered as a fixed surface (two-dimensional) temperature. This is T_0, a constant temperature along all the outer surface of the base plate (that is the surface that would be in contact with the chip in real life application). Figure 10 shows it in a visual way.

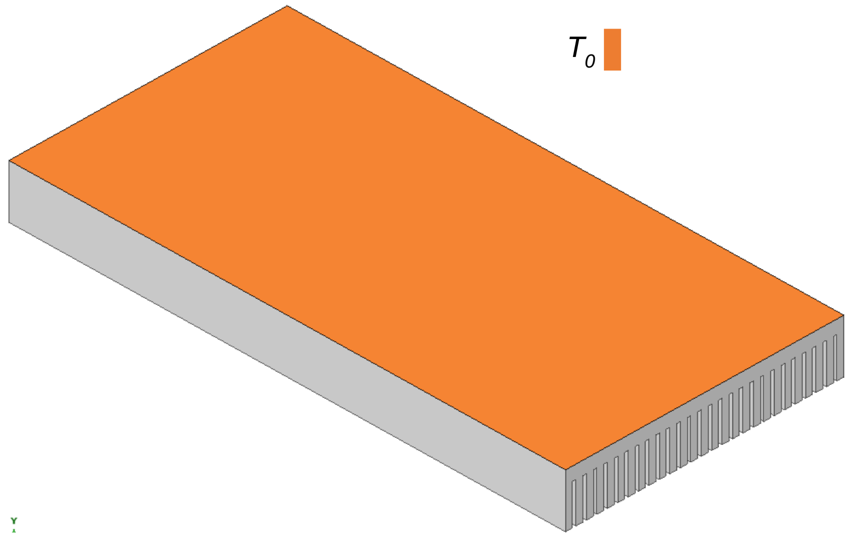


Figure 10: Visual representation of fixed temperature at the base plate (T_0)

The explanation to this simplification is resumed in the following paragraphs:

There are two options to represent the heat that the chip transfers to the base plate. One is fixing a constant heat flux that crosses perpendicularly the outer surface of the heat base plate in direction to the fins and the other option is to fix a constant temperature at outer side of the base plate.

Since it is interesting to fix a constant temperature because the values of chip failure are known, as it is showed in Figure 11 it can be more useful to choose the second option.

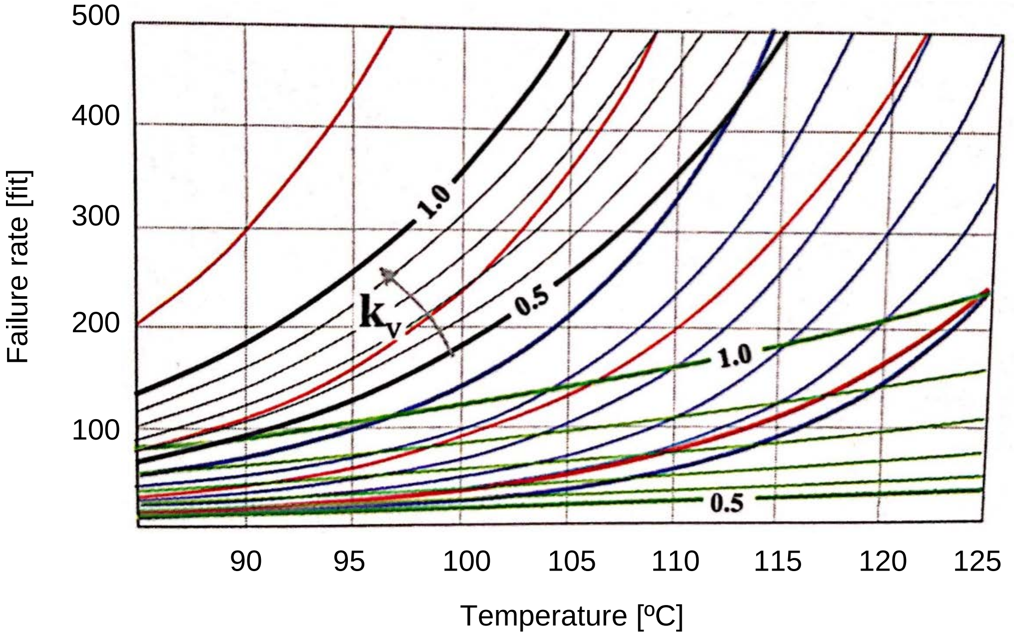


Figure 11: Failure rate of electronic components (Example: condensers). *Source: (7)*

As it can be observed in Figure 12, there are two configurations of temperature distribution, each one for each option: constant heat flux (left) or constant temperature (right). It is known that different thermal boundaries lead to different thermal resistance expressions (1), but according to the work done by Knight (9), both expressions procure similar results. The temperature distribution at the base plate when there is a constant heat flux applied is so uniform that it can be simplified as a constant temperature.

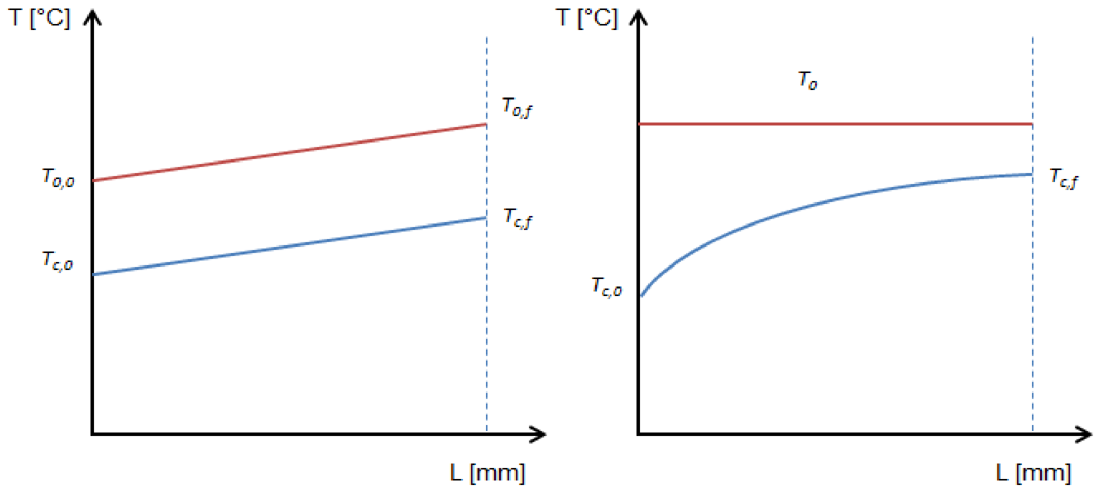


Figure 12: Evolution of temperatures with constant heat flux (left) and constant Temperature (right). *Source: Own illustration*

Figure 12 shows the temperature profile of both different thermal configurations along the length of the heat sink. As it can be seen, T_0 affects the curve of the coolant´s temperature (T_c). However, for practical purposes, the difference is negligible, so it can be chosen constant T_0 as the thermal boundary.

- Fixing ΔT_{max} to 20°C: As defined in 4.3, ΔT_{max} is the difference between the temperature of the outer side of the base plate and the temperature of the fluid coolant at the inlet ($\Delta T_{max} = T_0 - T_{c0}$). Fixing this boundary condition it is a simplification because it ignores the fluid flow temperature at the outlet (T_{cf}). The correct way to evaluate this temperature difference is with the logarithmic mean temperature difference, which is used in average heat exchanger calculations (8)[Equation 24].

$$\mathrm{LMTD}=\frac{\Delta T_A-\Delta T_B}{\ln\left(\frac{\Delta T_A}{\Delta T_B}\right)}$$

[Equation 24]

Where LMTD is the logarithmic mean temperature difference and ΔT_A is the difference between the hot streams and ΔT_B beteween the cold streams of the heat exchanger. However, in heat sink calculations ΔT_{max} is commonly used as the temperature difference to calculate thermal resistance (1).

With that said, it was a carefully chosen criteria to fix ΔT_{max} to 20°C so that the temperature increasing of the fluid would have only been of a few degrees so that not including T_{cf} in R_{th} calculation would not have influenced

- Fluid properties at T_0: One of the main simplifications was to evaluate the properties of the fluid coolants at the temperature that they had at their inlet. In common heat transfer calculations, the fluid is evaluated at its average temperature (T_m). The average temperature of the fluid, in the case concerning this thesis, would be obtained with [Equation **25]**.

$$T_m = \frac{T_{c0} + T_{cf}}{2}$$

[Equation 25]

That would have meant that a hypothesis for estimating an initial value of the outlet flow temperature (T_{cf}). Then, successive iterations of the same calculation should have been made in order to achieve a convergence of T_{cf}.

Nevertheless, as the temperature difference between T_0 and T_{c0} is just 20°C, it was estimated that T_{cf} and T_{c0} would only differ by a number comprised in a range that would go from 0,5°C to 3°C. With just a few degrees centigrades difference, the evaluation of the properties can be done at T_0 without making a determinant influence in the solution. In this case, T_{c0} and T_m would only differ in a range of [0,25, 1,5] which is so small that it can be considered T_{c0} as the temperature to measure the coolants properties.

- Other heat transfer considerations: finally, before ending this section, a couple of points have to be mentioned. The first of them is that the materials were considered isotropic, so that the heat transfer was uniform in all directions. The second is that the fins were considered rectangular beause that allows an easier analysis referred to the geometry. Finally, no overflow was considered above the fin tip, that is why the fluid domain height is the same as the fins. It is proved that an overflow beyond the fin tip decreases the efficiency of the heat transfer (1).

4.4.5 Values for the constant parameters

The next table [Table 2], shows the values that were set for the constant parameters.

Concept	Description	Value	Units
H	Fins height	6	mm
L	Length of the heat sink	80	mm
W	Width of the heat sink	40	mm
d_{bp}	Thickness of the base plate	2	mm
T_0	Temperature of the outer surface of the base plate	85	°C
T_{c0}	Inlet temperature of the coolant	105	°C

Table 2: Values for the constant parameters of the system

As it can be observed the geometrical parameters were chosen in order to have an average and representative heat sink in Power Electronics applications.

On the other hand, T_0 is set to 105°C, a temperature that is in the limit for a chip to have failure issues. Therefore, and taking into account that ΔT_{max} is 20°C, T_{c0} is automatically set to 85°C.

The setting of the variable values is explained in 6.1.1.

4.4.6 Steps for solving each system configuration

Finally, as an end to this chapter, it is of interest the steps that were followed for obtaining the solution by means of the analytical method.

First of all, the thermal properties of the coolants (conductivity, density, specific heat and viscosity) were evaluated at T_0 as explained in 3 and shown in Table 3. Each fluid coolant had their properties calculated with their own expressions [Equation 1 to Equation 8] for obtaining their particular thermal and mechanical values.

Secondly, with the material properties and the constants and variables already set, Equation 13 and Equation 15 were used to obtain Reynolds number and Prandtl number. In the same way, the correlation for Nusselt number written by Dittus-Boelter and represented in Equation 16 was also evaluated. Parallel to this process, the friction coefficient was also obtained using Equation 18.

Once the non-dimensional numbers were obtained, it was time to calculate the thermal resistance of the surface with Equation 12. Then, knowing R_{th} and ΔT_{max} and applying Equation 20 the thermal dissipated by the system was procured. Subsequently, the outlet temperature of the fluid coolant (T_{cf}) was calculated using Equation 19. Analogously, Equation 21 permitted the obtaining of the pressure drop (Δp).

Once arrived to this point, the process of calculation was finished. Both parameters for being compared were known: T_{cf} and Δp. So it is time to jump to the simulation model of calculation.

Coolant	T_0 [ºC]	$c_{pc}(T_0)$ [J/kgK]	λ_c(T0) [W/mK]	$\rho_c(T_0)$ [kg/m^3]	$\upsilon_c(T_0)$ [m^2/s]
Liquid water	85,00	4183,00	0,650	1000,00	$2{,}40 \cdot 10^{-7}$
Water-glycol	85,00	2675,00	0,260	1071,50	$2{,}95 \cdot 10^{-6}$
Galden HT-135	85,00	962,96	0,0650	1720,00	$4{,}80 \cdot 10^{-7}$
Novec 7500	85,00	1218,00	0,057	1489,00	$3{,}68 \cdot 10^{-7}$
Fluorinert FC-3283	85,00	1146,00	0,060	1669,00	$3{,}30 \cdot 10^{-7}$

Table 3: Value of the physic properties for each fluid coolant

5 Simulation method

As it is explained in 2.1, the main objective of the thesis involved proving the analytical method with a simulation method. This last method required the usage of a solving software that could predict the real values of the system the most accurate as possible. As it is described by its developers themselves, ANSYS Fluent software is the most powerful computational fluid dynamics (CFD) tool available (10). It was a matter of interest to obtain the results with this tool neither for adding more information to the previous studies that Dr. Guan nor Prof. Dr. – Ing. März has done.

The following sections explain the process of set-up for ANSYS Fluent and the solving methods used.

5.1 ANSYS Fluent: the software

ANSYS software is a generalised model of solving the heat transfer issue. That means that applies the equations of mass conservation, energy and quantity of movement to a differential of fluid volume. Doing that, a system of equations can be found and then the energy transmitted in the form of heat can be deducted (8).

It is a numerical way of solving where a series of discrete volumes of control are defined (cells) and the system of equations is solved in each of them sequentially. For this, it is needed the definition of initial and boundary conditions.

ANSYS software allows the user to import the geometry from a CAD platform or even design it by their own with the option of *Design Modeller*. Moreover, automatically meshes can be created with ANSYS meshing software. It has the tools to create, modify and improve (in terms of quality) specifically meshes for fluids. It has also different solvers that can accurately predict the behaviour of the fluid depending on the type of flow that is being treated. Finally, it has powerful post-processing options that allow the user to get the highest performance from the results.

5.2 ANSYS Fluent: configuration

For a correct and valid numerical simulation, the configuration of the parameters that are involved has to be in a way that allows the simulation to be as closest to reality as it can be. That means, that from the geometry building to the solver setting, the choice of features needs to be coherent and well argued. The following sections discuss the building of the simulation model.

5.2.1 Geometry

Figure 13 shows one of the geometries used for the ANSYS simulations. As it can be observed, it is a very simple construction, with the constant parameters fixed with the same values as it has been explained in 4.4.2, so that the geometry of the simulation model and the analytical model were coincident.

The fluid domain is shown in blue and it is just the space between the fins and the base plate. It can be observed that the bottom boundary of the flow is at the same height as the fins, so that there is no overflow. In grey is represented the heat sink (both base plate and fins).

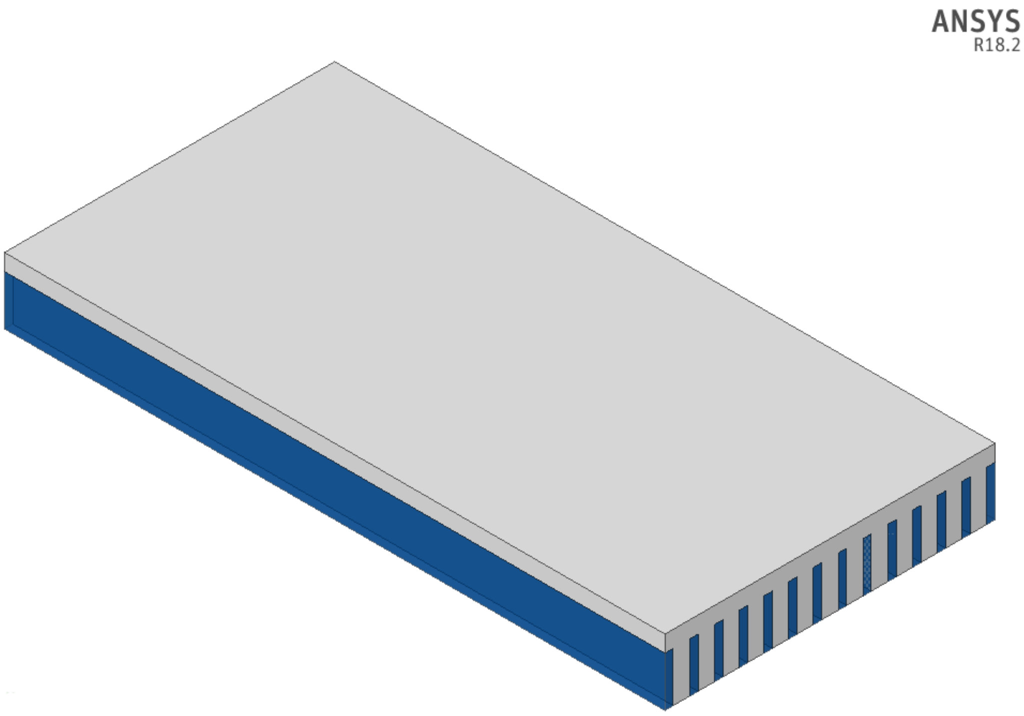


Figure 13: geometry of the system domain created for ANSYS simulations.

ANSYS allows the user to define components that later will have an influence on the set-up of the simulation. Thus, five components of the system were defined:

- Inlet and outlet: Figure 14 shows the inlet (analogously is the outlet defined) definition. Where the fluid flow enters the system. It is the boundary surface of the fluid domain on the front face of the volume domain.

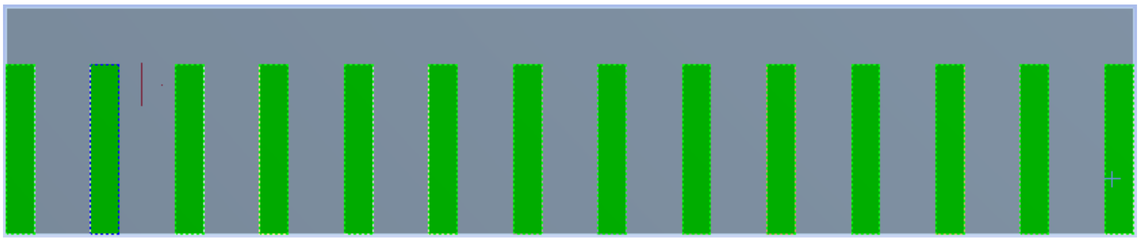


Figure 14: Inlet surface from where the flow enters the system (in green)

- Chip: in the same way as the temperature of the base plate (T_0) is defined as a thermal boundary condition (4.4.3) in ANSYS the chip is defined as the outer surface of the base plate, so that once in the set-up mode, it can be applied the boundary condition required [Figure 15].

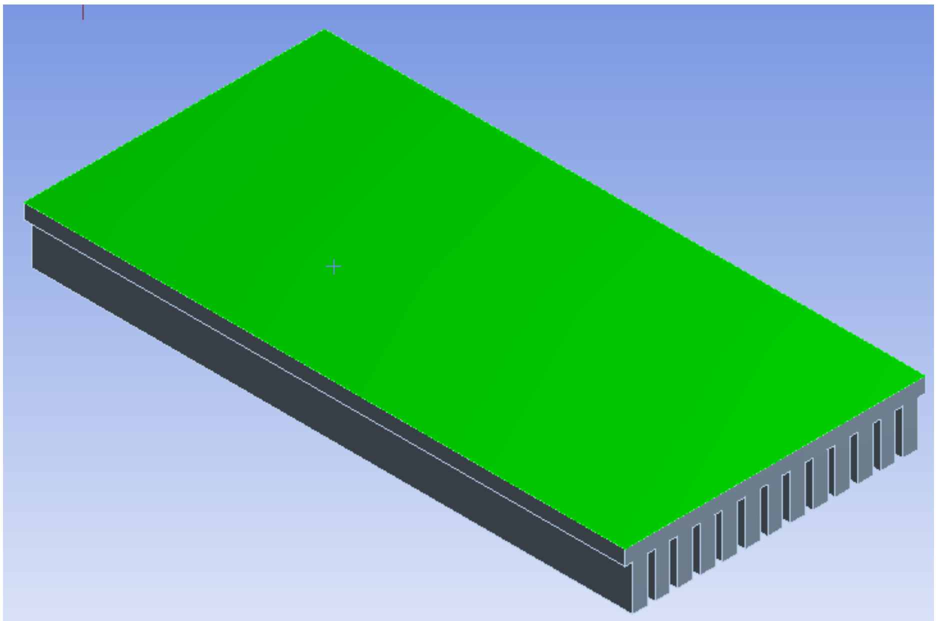


Figure 15: The chip represented as the outer surface of the heat sink´s base plate (in green)

- <u>Heat sink and fluid domain</u>: Finally, the heat sink and the fluid domain are equally defined as volume components with the aim of applying the correct boundary conditions and interactions during the set-up.

5.2.2 Mesh

The next step was setting the mesh. About it, in the Table 4, there is a resume of the main features of it.

Mesh details	
Number of nodes	ca. 420000
Number of elements	ca. 400000
Type of elements	Hexahedral
Type of mesh	Structured
Inflation	Yes
n° of inflation layers	10
First layer thickness	0,05 mm

Table 4: Definition of mesh features in ANSYS

The number of nodes and elements was chosen in function of the maximum the ANSYS Academic version allows the user to define. So, with a maximum of 512000 cells allowed, the models were designed with around 400000. It was considered enough as it is not a complex geometry for the solver. The element type hexahedral [Figure 16] was selected as it saves more time and computational energy because of the fewer amounts of nodes involved in the system. It also makes the mesh structured, so easier and quicker to solve by the software. There were also included inflation layers in the fluid domain, with the aim of obtaining the most accurate behavior possible of the fluid flow near the walls of the heat sink [Figure 17].

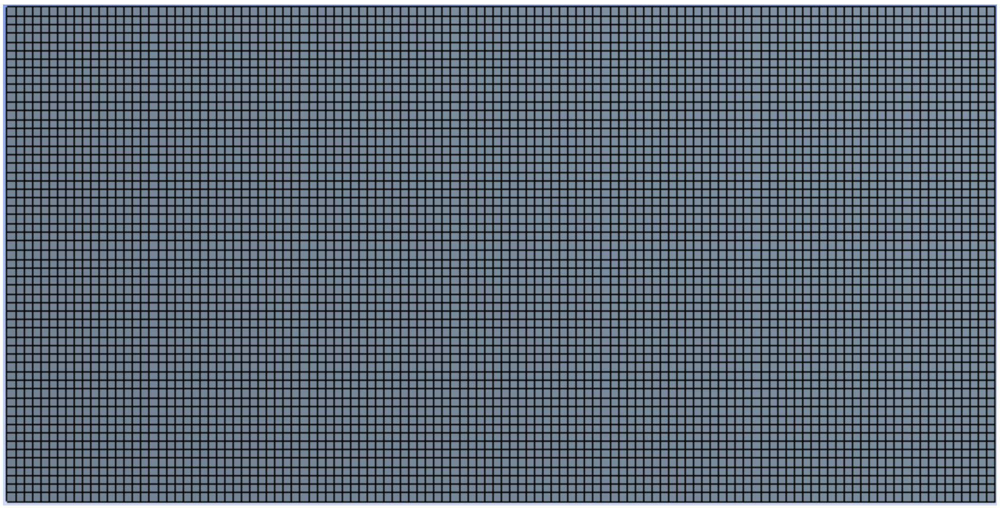


Figure 16: Top view of the hexahedral mesh

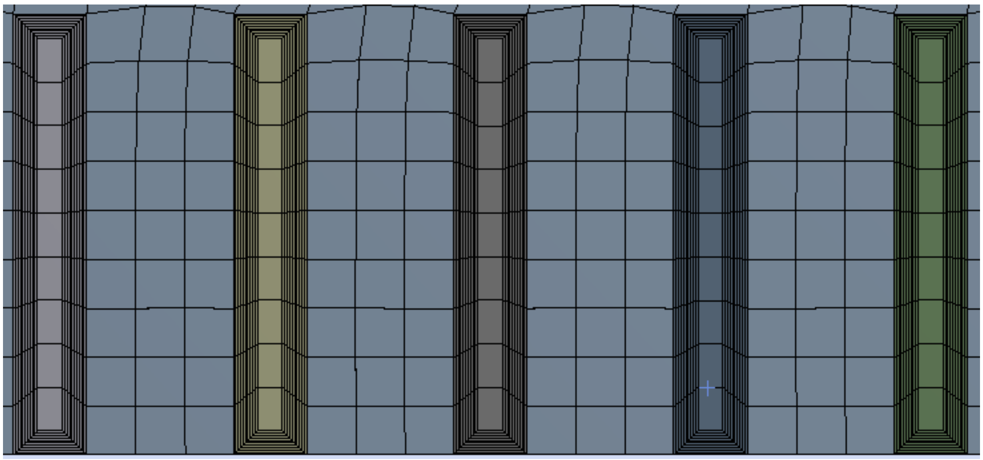


Figure 17: Detailed front view of the inflation layers in the fluid domain.

5.2.3 Set-up

After the mesh is designed, the program itself leads the user to set the boundaries and restrictions to which the domain is going to be subjected. Table 5 summarizes the parameters that had to be defined in each section for making the solving process as accurate to reality as possible.

Firstly, the solver was set to pressure based. This is the recommended configuration when the fluid has low Mach number (11). In other words, when the flow is subsonic and acts as incompressible. This happens when Mach < 0,3, which it is true in all the cases studied in this thesis. Pressure-based solver saves computational time as it assumes density to be constant hence equations are computed based on pressure equations. On the other hand, in density-based solver, density is a variable which requires a new equation to be solved.

In the same section, the steady option was activated in order to have a non-time dependant solution (the operation in an average situation is what is searched).

Secondly, in the Models section the energy equation was enabled in order to allow heat transfer between interfaces. The viscosity model was set to standard *k-omega.* This is a two-equation Reynolds-averaged Navier-Stokes model that pairs the turbulent kinetic energy (k) with the specific rate of dissipation of kinetic energy (*omega*). It is a popular model between engineers, and can predict near-wall interactions more accurately than other ones (12). Although it can be a complex choice, it was elected as the one that could most accurately predict the turbulence in the cases studied.

Next step was defining the materials. Each new coolant had to be introduced from zero to the program with their required properties evaluated at T_0. The solid material was set as aluminium and the properties were set the same as the ones in the analytical model with the aim of having the same conditions. Logically, the fluid material was assigned to the fluid domain in each case and the solid to the heat sink cells.

Finally, the boundary conditions had to be defined. As it can be observed in Table 5, they are the same as the ones introduced in 4.4.3 for the analytical method. The outer base plate wall was set at a constant T_0, the fluid inlet surface was defined as a velocity inlet (u_c) that has an initial temperature of T_{c0} and a hydraulic diameter (for the prediction of turbulence) correspondent to the fluid domain $D_h=\frac{4d_fH}{2(d_f+H)}$. The *Wall heat sink / coolant* refers to the surface in contact with both fluid domain and heat sink. Therefore, it was set to be coupled, so that convection effect could be calculated through it. As an end to this section, the system boundary walls, that is the walls that are in contact with the environment that surrounds the system studied were set to be adiabatic, so that the system remained isolated from external inputs or outputs.

Section	Parameter	Defined as
General	Solver	Pressure based
	Time	Steady
Models	Energy	On
	Viscous	Standard k-omega
Materials	Fluid	Current coolant
	Solid	Aluminium
Boundary Conditions	Outer base plate wall	Constant temperature (T_0)
	Fluid inlet	Velocity inlet (u_c) Temperature inlet (T_{c0}) Hydraulic diameter (D_h)
	Fluid outlet	Pressure outlet
	Wall Heat sink / coolant	Coupled walls - Convection
	System boundary walls	Adiabatic walls

Table 5: Set-up parameters for ANSYS Fluent

5.2.4 Solution and post-processing

Once the set-up is configured, ANSYS software offers a way of solving based on iterations until the solution converges. For achieving this, ANSYS has a default criterion based in scalar residuals which was the one used. Usually, in between 100 to 200 iterations the cases were solved but in some of them under-relaxed coefficients had to be used for making the convergence possible.

ANSYS post-processing methods give the option of obtaining the results in different ways. Thus, it permits the creation of multiple visual graphics that help with the analysis. In particular, for this thesis the specific tool of *area-weighted average* was very useful. It gives the average value of the magnitude selected along a surface. It was used to obtain the average values of temperature and pressure at the inlet and at the outlet.

Arrived to this point, the theoretical explanation of the thesis is finished. The process of solving by the analytical method firstly and by the numerical method (computer simulations) secondly have been explained. Now, it is time to follow the process of the obtention of the results, the analysis and the final conclusions obtained.

6 Results

This chapter shows the results that were obtained and the analysis that was done afterwards. These results are all procured in the way that it has been explained in the previous chapters (4 and 5). However, firstly it is needed to explain the sequence followed for the obtaining of them.

As it was discussed in section 2.2, the first objective was to validate Dr. Guan´s theoretical model with ANSYS simulations. Therefore, the first fluid that was analysed was liquid water. If the results were satisfying, this would permit the advance to other different coolants.

6.1 Initial calculation: liquid water

For a successful study of liquid water as a fluid coolant a method which later was repeated with the other fluids was used. It consisted in the analysis of five cases in which the variables defined in 4.4.2 (d_f, s_f, u_c) where set with different values in each one of them. Having these five different configurations of the system and once compared them by the analytical and the numerical method, it would be enough to have the certainty to accept or refuse the analytical model.

6.1.1 The five cases studied

The difference between one case and another lies in the value of the variables. Thus, in each case the variables are changed so different configurations are set and the comparison between analytical and numerical methods is valid and useful.

As defined in 4.4.2, the $P_{hyd,}$ although it is not an entry variable, it has a lot of importance in the setting of the cases and defines the inlet velocity, therefore the volumetric flow. So, for starting the variable assignment a P_{hyd} equal to *0,33W* was defined. It is an average value for the hydraulic power in Power Electronics cooling cycles (1). In the same way, fin thickness and distance between fins were defined to represent average heat sink configurations in Power Electronics. Table 6 summarizes the definition of the variables.

Fluid coolant: liquid water		**Values**				
Concept	**Abbreviation**	**Case A**	**Case B**	**Case C**	**Case D**	**Case E**
Hydraulic power	P_{hyd} [W]	0,33	0,33	0,33	0,33	0,33
Distance btw. fins	d_f [mm]	2,50	0,50	1,00	1,50	2,00
Fin thickness	s_f [mm]	1,00	1,00	2,00	3,00	5,00
Velocity	u_c [m/s]	1,91	1,37	1,81	2,11	2,47

Table 6: Value of the variables in the different cases with liquid water as a fluid coolant.

As it can be observed in Table 6, the hydraulic power is the same for each case. That, together with the geometry of the heat sink automatically defines the inlet velocity of the coolant flow

[Equation 22] and [Equation 23]. Both two geometrical values (d_f and s_f) were also set to average and different values as it can also be seen.

6.1.2 Analytical results

Up to this point, everything was defined so that it was possible to proceed to the solution. Firstly, with the analytical procedure and secondly, using ANSYS simulation software.

Following the process of calculation defined in 4.4 and the variables in Table 6, the following results were obtained:

Analytical results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**ΔP [mbar]**
Liquid Water	A	28020,00	85,00	86,31	**1,31**	**10,09**
	B	5255,00	85,00	89,41	**4,41**	**30,13**
	C	12930,00	85,00	88,06	**3,06**	**22,75**
	D	21100,00	85,00	87,43	**2,43**	**19,52**
	E	30870,00	85,00	87,14	**2,14**	**19,45**

Table 7: Results of the analytical calculation with liquid water as a fluid coolant.

Table 7 shows the parameters of interest (final temperature of the fluid and pressure drop) concerning the analytical solution. Although without the simulation solution the model cannot be validated yet, in a first look it can be noticed that the case that dissipates more heat (so the most interesting one from the heat transfer point of view) is *B*. It is the one that has the biggest temperature increasing, which is directly proportional to the thermal power dissipated.

The reason of this resides in the small values of fin thickness (s_f) and distance between fins (d_f). This means that a high number of fins form the heat sink, therefore a high value of the surface of convection [Equation 10]. On the other hand, because of this same reason, a biggest area of contact between solid and fluid leads to a higher value of the friction coefficient, which makes the pressure drop to be increased. As it can also be seen in Table 7, case *B* is the one with the value of pressure drop ΔP over 30 mbar.

These early conclusions just advance one of the issues concerning the choice of the geometrical dimensions of heat sinks. It is a matter of preference which type of heat sink wants the engineer to choose. The optimization can be done in different ways: either in heat dissipation terms or in minimizing the pressure drop but also in material volume or in volumetric flow. The choice of design has always to be done taking into account the application for which the heat sink is going to be designed and the restrictions to which is subjected.

6.1.3 Simulation results

Analogously to section 6.1.2, the simulation results were obtained following the process of calculation explained in 5.2. They are the following:

Simulation results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**ΔP [mbar]**
Liquid Water	A	28020,00	85,00	86,33	**1,33**	**10,99**
	B	5255,00	85,00	90,01	**5,01**	**40,48**
	C	12930,00	85,00	88,28	**3,28**	**26,69**
	D	21100,00	85,00	87,75	**2,75**	**24,47**
	E	30870,00	85,00	87,49	**2,49**	**20,48**

Table 8: Results of the numerical calculation (ANSYS) with liquid water as a fluid coolant.

Which in a quick view Table 8 shows a slight difference between the values highlighted in bold (*ΔTc and ΔP*) compared to Table 7. This is the difference that has to be analysed.

ANSYS lets the user obtain visual and graphic results. It is interesting to analyse it from that point of view too. Therefore, the behaviour of all the points of the coolant as it goes through the fluid domain can be easily observed. Figure 18 is a good example of it.

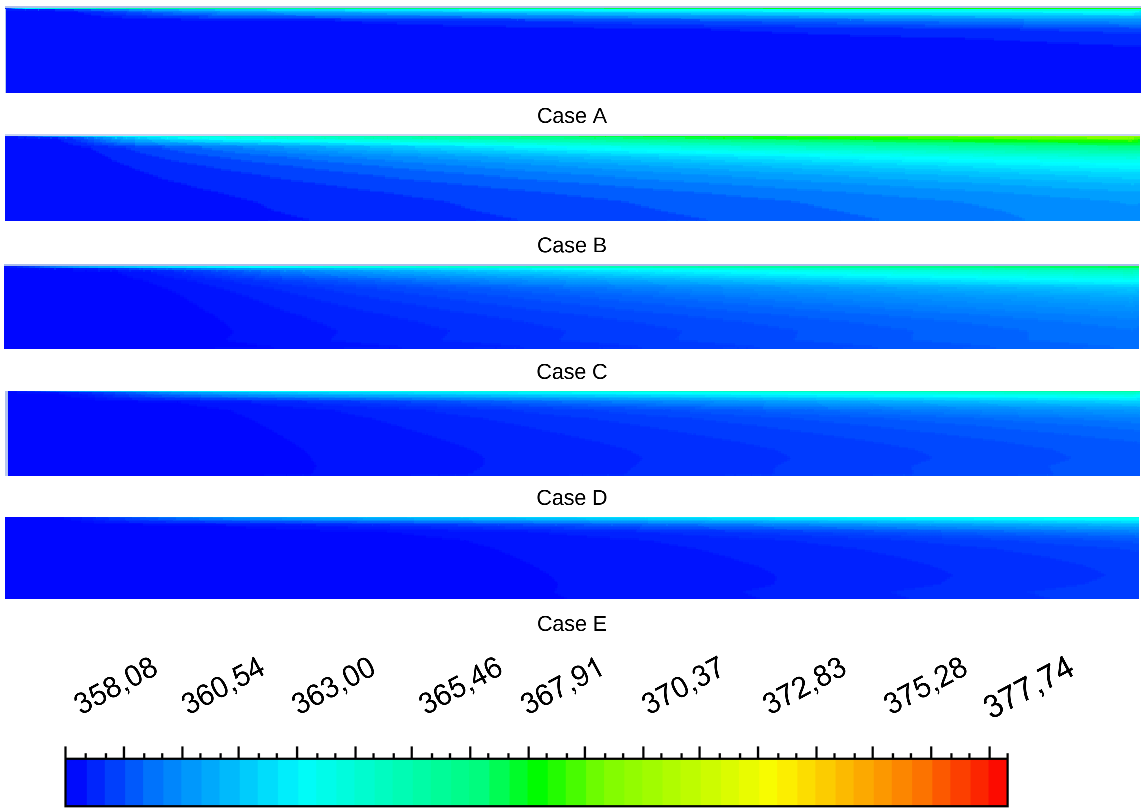


Figure 18: Visual representation of the temperature evolution of the water flow on a longitudinal plane situated in the middle between two fins. Temperature given in Kelvins *Source: Done with ANSYS software.*

There, it can be seen the fluid temperature evolution in a longitudinal vertical plane situated in the centre of the fluid domain between two fins. The left side would be the inlet and the right side would be the outlet.

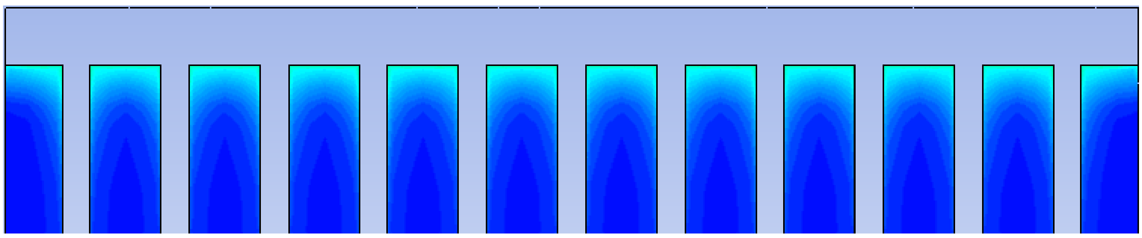
Comparing it to the results in Table 8, it can be observed that Case B is the one in which the temperature suffers the most of the change and Case A the one with the least. These results are coherent with the values of temperature difference shown in Table 8.

The contour post processing tool allows seeing the way the heat transfer is produced. Observing Figure 18, it can be known how the convection process works. From the inlet to the outlet, the fluid flow is subject to a continuous increasing of temperature from the top to the bottom. In Case A, the bottom flow (the one that is in contact with an adiabatic wall) at the outlet it even remains with the same temperature as at its inlet and in Cases D and E it is close to remain intact. This longitudinal distribution of temperatures is better understood with the transversal contour of temperatures at the outlet [Figure 19].

It is interesting to see how the temperature is distributed at the outlet of every channel. Logically, the highest temperatures of the fluid coolant are at the top, where the temperature difference is bigger. It can also be observed how in Case B or Case C the whole surface is affected by the temperature increasing. On the other hand, Case A and Case E are barely affected at their bottom by the temperature increasing. The enhanced heat transfer produced by the fins it is also visible: the temperature of the fluid near the walls is always higher than the temperature at the centre zone. That is why the isothermal layers that form the temperature distribution have their particular convex-type shape.

Referring to the temperature calculation, the inlet one can be easily obtained because it is uniform all over the surface. The opposite happens with the outlet temperature which it is different at every point as Figure 19 shows. Which one should be selected? It is crucial to pick the appropriate one for developing valid conclusions afterwards. The average temperature at the outlet is the one that it is searched and fortunately, ANSYS post-processor has an *Area-weighted average* tool that gives the average value of a variable along a surface. So, applying this tool to the temperature variable at the outlet surface is how the values of T_{cf} in Table 8 are obtained. Analogously, it is done with the pressure drop at the inlet and at the outlet. With the help of this tool values in Table 7 and values in Table 8 can be compared.

The next section (6.1.4) shows this comparison and analyses whether the analytical hypothesis should or not be rejected.



Case A

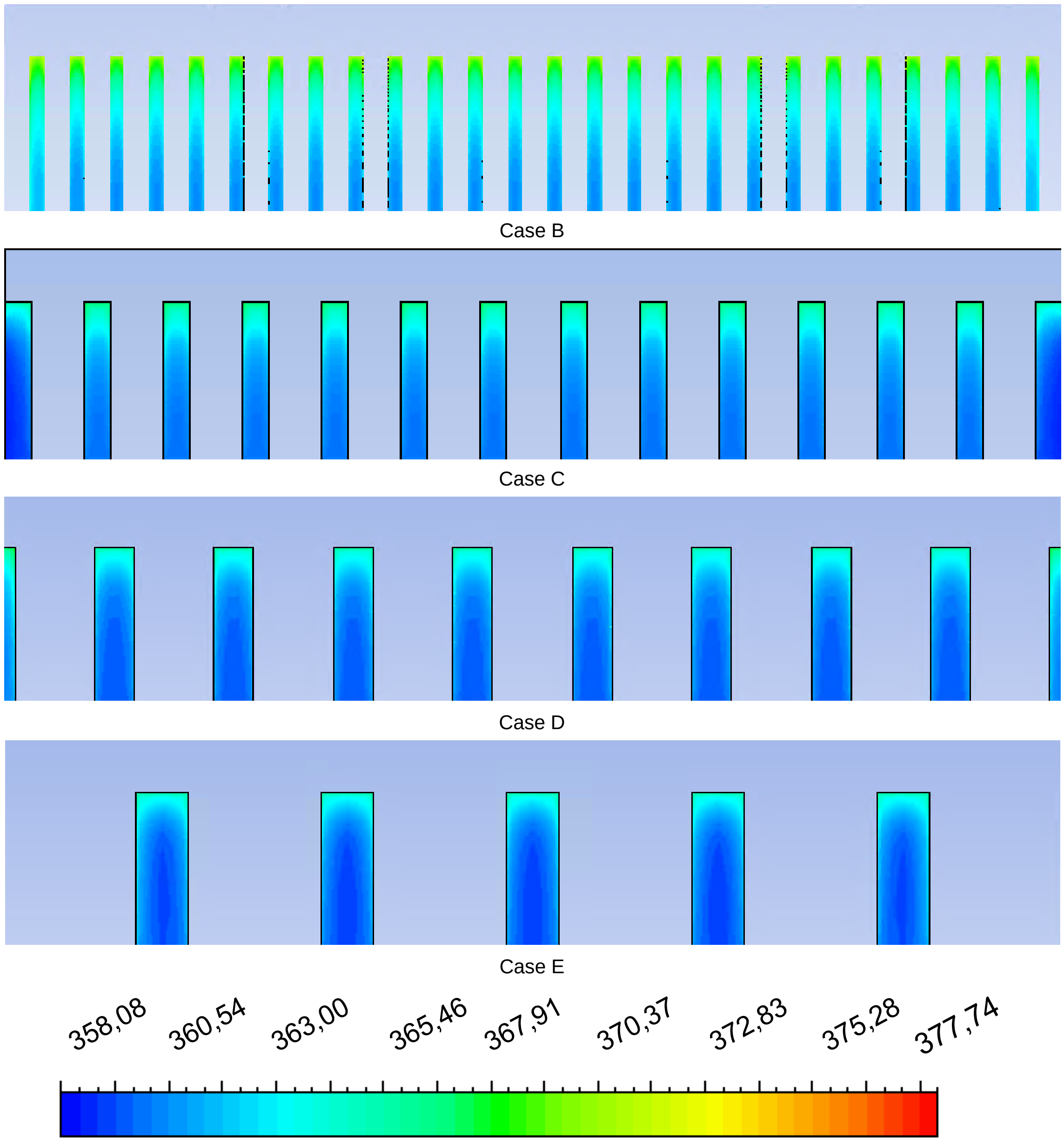


Figure 19: Transversal contour of fluid domain temperature at the outlet. *Done with ANSYS software.*

6.1.4 Comparison and conclusions

Table 9 shows the comparison between the analytical values and the simulation ones. It can be observed that the average difference is only 9,21%, less than 10% which is a value that validates the hypothesis. Less than 9-10% would be unrealistic because of the differences that may always exist between the two methods due to unavoidable error margins. The pressure drop has also a low difference value, although it is not as low as the one for the temperature difference.

		Analytical		Simulation		Difference [%]	
Coolant	**Case**	**ΔT_c [ºC]**	**Δp [mbar]**	**ΔT_c [ºC]**	**Δp [mbar]**	**ΔT_c**	**Δp**
Liquid Water	A	1,31	10,09	1,33	10,99	**1,58%**	**8,27%**
	B	4,41	30,13	5,01	40,48	**12,00%**	**25,57%**
	C	3,06	22,75	3,28	26,69	**6,80%**	**14,76%**
	D	2,43	19,52	2,75	24,47	**11,64%**	**20,23%**
	E	2,14	19,45	2,49	20,48	**14,06%**	**5,01%**
					Average	**9,21%**	**14,77%**

Table 9: Comparison between analytical and simulation methods for liquid water as a fluid. Difference calculated respect to the ANSYS values.

With the objective of having a better idea of the difference between the two types of calculation, it is better to show it graphically with the ANSYS results superimposed on the analytical ones in a plot. For that, it is defined a new parameter called COP_{hyd}. This non-dimensional parameter is a way of calculating the heat dissipation efficiency respect to the hydraulic power used. That is: $COP_{hyd}=\frac{P_{th}}{P_{hyd}}$ [Equation 26] [Figure 20].

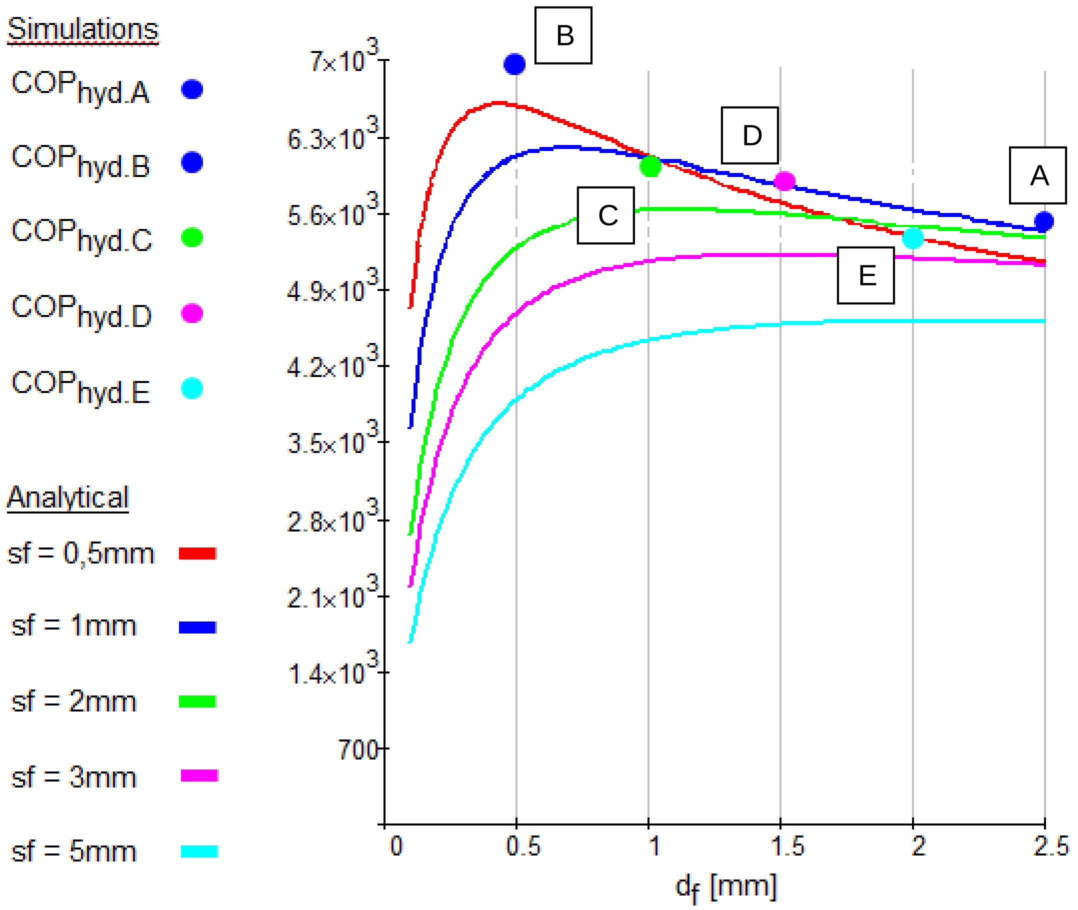


Figure 20: Graphical comparison between analytical and numerical results (liquid water). Done *with MathCAD software.*

Figure 20 is a visual representation of the COP_{hyd} for the five cases studied (A, B, C, D and E). The curves represent the analytical method calculated with MathCAD software. The advantage of this tool is that allows the user to plot a series of consecutive points easily. That is why it is not complicated to draw the curves for any parameter that is needed. It can be observed that the COP_{hyd} has been plotted as function of the distance between fins (d_f) and at the same time it has been divided into five values for the fin thickness. In that way both variables, which affect significantly the heat dissipation, have been represented at the same time.

The five points painted in different colours represent the five cases for the simulation method (note that the red curve does not have any point assigned). As it can be observed in the box legend, the colours of the points are paired with the colours of the analytical method curves. Then, points A and B have to be related to the blue curve, C to the green one, D to the pink one and E to the cyan one. It can be seen that the difference described in Table 9 is represented by the vertical line that unites each point with its same-coloured line.

Once realised that, it can be checked that case A is the one with less visual difference as well as the one with less % difference (just 1,58%). C comes next (6,80%) and afterward B, D and E are the ones with the most space between them. The values in the graphic are directly proportional to the ones in Table 9. A conclusion that can be extracted after observing Figure 20 is that although it exists a difference (in liquid water case it is quite small) the behaviour of it is coherent and uniform. That means that looking to the graphic, it can be noticed that the ANSYS values are always above the analytical values with similar distance difference from each one them to their respective curves.

After having seen the close values of the result comparison, the conclusion is that the initial hypothesis for the analytical model is valid. The Dittus-Boelter expression [Equation 16] is useful for the conditions studied in 6.1 as well as it is the one for the friction factor [Equation 18]. Both can make an accurate prediction of the thermal and mechanical behaviour of the heat dissipation with liquid water through a plate-fin heat sink. This is a reaffirmation of Dr. Guan´s studies around the thermal behaviour of plate-fin heat sinks.

As it was defined in the main objectives of the thesis, this first calculation was a touchstone for the next ones. So, once liquid water as a fluid coolant has been satisfactorily tested it is time to proceed with the investigation and test the next fluid coolants.

6.2 2nd calculation: water-glycol

Following liquid water, ethylene glycol or, popularly called, water-glycol had to be tested as a coolant. In the same way as in 6.1, five cases were set. It was wanted that the variables kept the same value as in liquid water so that it could be seen how different coolants worked subjected to the same conditions. Therefore, d_f, s_f, u_c were defined with the same value as in Table 6. Nevertheless because of a different density, the hydraulic power used to move the flow changes. The new table of variables for ethylene glycol is the next one:

Fluid coolant: water-glycol		Value				
Concept	**Abbreviation**	**Case A**	**Case B**	**Case C**	**Case D**	**Case E**
Hydraulic power	P_{hyd} [W]	0,66	0,66	0,66	0,66	0,66
Distance btw. fins	d_f [mm]	2,50	0,50	1,00	1,50	2,00
Fin thickness	s_f [mm]	1,00	1,00	2,00	3,00	5,00
Velocity	u_c [m/s]	1,91	1,37	1,81	2,11	2,47

Table 10: Value of the variables in the different cases with water-glycol as a fluid coolant.

Where, as it can be seen in Table 10 the only thing that differs from Table 6 is $P_{hyd.}$ It continues to be constant as it was wanted to be defined firstly but it has a new value which is also conceivable in cooling circuits for Power Electronics.

In the same way as with liquid water, the analytical and the simulation results were obtained. Table 11 and Table 12 represent the results for the analytical and the simulation methods respectively.

Analytical results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**Δp [mbar]**
Water-glycol	A	2285,00	85,00	85,68	**0,68**	**20,23**
	B	428,56	85,00	88,05	**3,05**	**60,44**
	C	1055,00	85,00	86,72	**1,72**	**45,62**
	D	1720,00	85,00	86,20	**1,20**	**39,13**
	E	2517,00	85,00	85,93	**0,93**	**39,01**

Table 11: Results of the analytical calculation with water-glycol as a fluid coolant.

Simulation results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**Δp [mbar]**
Water-glycol	A	2285,00	85,00	86,18	**1,18**	**29,61**
	B	428,56	85,00	89,94	**4,94**	**176,94**
	C	1055,00	85,00	87,68	**2,68**	**83,14**
	D	1720,00	85,00	87,10	**2,10**	**78,34**
	E	2517,00	85,00	86,63	**1,63**	**54,64**

Table 12: Results of the simulation with water-glycol as a fluid coolant.

And with just a quick look it can be seen that the difference between analytical and numerical results seems quite big. The next table represents the comparison between them:

		Analytical		Simulation		Difference [%]	
Coolant	**Case**	**ΔT_c**	**ΔP**	**ΔT_c**	**ΔP**	**ΔT_c**	**ΔP**
Water-glycol	A	0,68	20,23	1,18	29,61	**42,20%**	**31,69%**
	B	3,05	60,44	4,94	176,94	**38,26%**	**65,84%**
	C	1,72	45,62	2,68	83,14	**35,90%**	**45,13%**
	D	1,20	39,13	2,10	78,34	**42,90%**	**50,06%**
	E	0,93	39,01	1,63	54,64	**42,82%**	**28,60%**
					Average	**40,42%**	**44,26%**

Table 13: Comparison between analytical and simulation methods for liquid water as a fluid. Difference calculated respect to the ANSYS values.

The percentage difference shows what it seemed firstly. The difference is high, nearly in a 50% in average. It also gets to 65% for the pressure drop in Case B. The reason for these results should be analysed in Table 11 and Table 12 where it can be observed that the Reynolds number is way lower than in the liquid water case. In fact, it is ten times inferior so the turbulence phenomenon decreases giving more space to a laminar flow.

If analysed visually with a graph as it was done with liquid water, the divergence between analytical results and simulations becomes clearer:

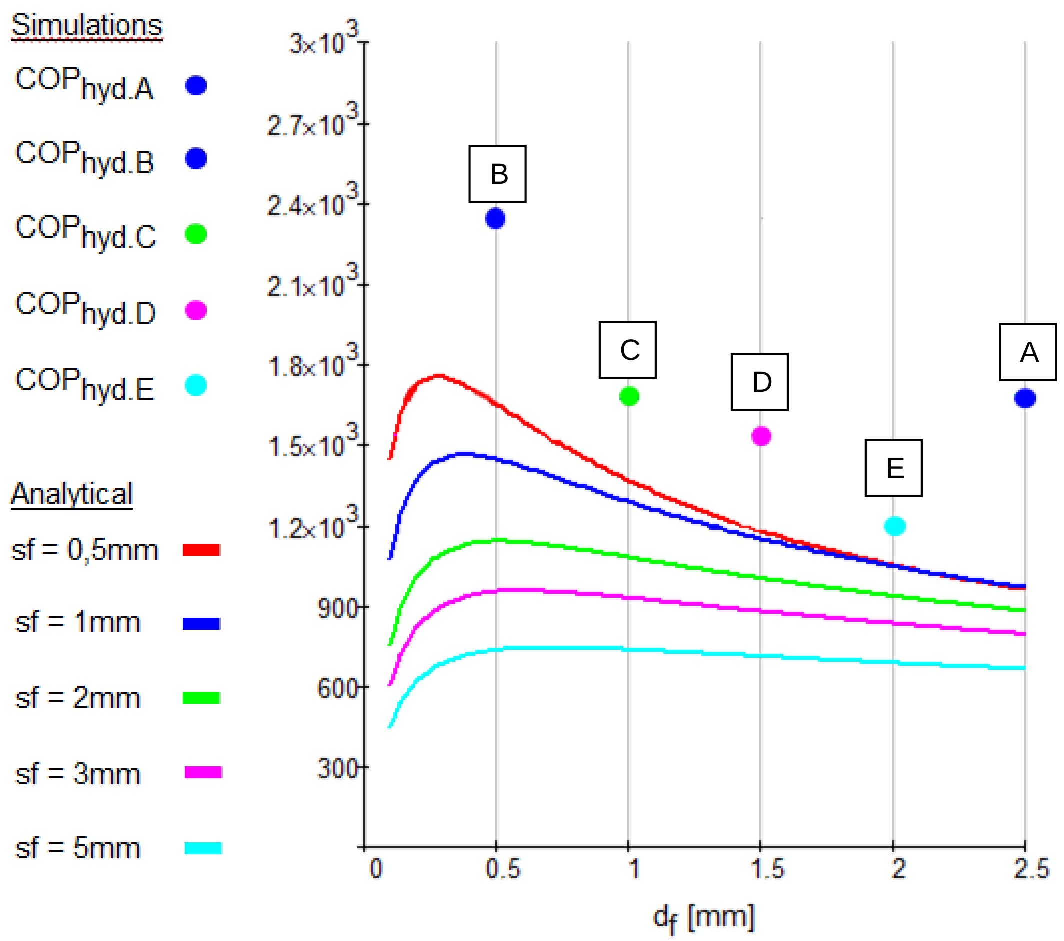


Figure 21: Graphical comparison between analytical and numerical results (water-glycol). *Done with MathCAD software.*

Figure 21 exemplifies the difference between the analytical and the simulation methods for water-glycol. Comparing it to Figure 20 (liquid water as the coolant) it can be observed that the vertical distance between the points (representatives of the ANSYS simulations) and their respective curves is significantly larger. Therefore, Dittus-Boelter equation cannot predict with the same accuracy heat dissipation with water-glycol as a fluid coolant as it does with liquid water. On the other hand, an interesting thing that can be seen is that the distribution of the simulation points in the plot although at a higher height respect the curves is kept the same as in liquid water. This can be taken as a reaffirmation of the coherent and uniform behaviour of the analytical method respect the simulations. ANSYS values are always above Mathcad ones and they *draw* a similar distribution as in liquid graphic [Figure 20].

However, these results led to an opportunity to prove the Reynolds influence on the validation of the analytical model. Therefore, for increasing Reynolds number the velocity was selected to be changed. That means that the hydraulic power should be increased to a higher value. This new value which was defined was of 600W, which is a totally unrealistic value for Power Electronics fluid cooling. Despite that, it was interesting to analyse it to know if the analytical model was valid for water-glycol in high Reynold numbers. Table 14 shows the new table of variables for each new case. It can be observed that the velocity has increased considerably, what makes these cases difficult to imagine in a real Power Electronics application. The distance between fins and fin thickness remain the same.

Fluid coolant: water-glycol		**Value**				
Concept	**Abbreviation**	**Case A'**	**Case B'**	**Case C'**	**Case D'**	**Case E'**
Hydraulic power	P_{hyd} [W]	600,00	600,00	600,00	600,00	600,00
Distance btw. fins	d_f [mm]	2,50	0,50	1,00	1,50	2,00
Fin thickness	s_f [mm]	1,00	1,00	2,00	3,00	5,00
Velocity	u_c [m/s]	22,71	16,29	21,58	25,14	29,43

Table 14: Value of the variables in the different cases with water-glycol as a fluid coolant.

The following three tables show the analytical and simulation results plus the comparison between them.

Analytical results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**Δp [mbar]**
Water-glycol	A'	27180,00	85,00	85,20	**0,20**	**1541,00**
	B'	5098,00	85,00	85,74	**0,74**	**4605,00**
	C'	12540,00	85,00	85,49	**0,49**	**3476,00**
	D'	20460,00	85,00	85,38	**0,38**	**2983,00**
	E'	29940,00	85,00	85,34	**0,34**	**2973,00**

Table 15: Results of the analytical calculation with water-glycol as a fluid coolant.

Simulation results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**Δp [mbar]**
Water-glycol	A’	27180,00	85,00	85,23	**0,23**	**1680,56**
	B’	5098,00	85,00	85,76	**0,76**	**6226,96**
	C’	12540,00	85,00	85,48	**0,48**	**4098,08**
	D’	20460,00	85,00	85,45	**0,45**	**3743,12**
	E’	29940,00	85,00	85,36	**0,36**	**3125,99**

Table 16: Results of the simulation with water-glycol as a fluid coolant.

		Analytical		**Simulation**		**Difference [%]**	
Coolant	**Case**	**ΔT_c [°C]**	**Δp [mbar]**	**ΔT_c [°C]**	**Δp [mbar]**	**ΔT_c**	**Δp**
Water-glycol	A‘	0,20	1541,00	0,23	1680,56	**13,63%**	**8,30%**
	B‘	0,74	4605,00	0,76	6226,96	**3,66%**	**26,05%**
	C‘	0,49	3476,00	0,48	4098,08	**0,70%**	**15,18%**
	D‘	0,38	2983,00	0,45	3743,12	**15,27%**	**20,31%**
	E’	0,34	2973,00	0,36	3125,99	**17,32%**	**4,89%**
					Average	**9,24%**	**16,61%**

Table 17: Comparison between analytical and simulation methods for liquid water as a fluid. Difference calculated respect to the ANSYS values.

As it can be observed in Table 15 and Table 16, the Reynold numbers of these new cases are in the same order as the ones that had the liquid water [Table 7]. Moreover, if we look at Table 17 the average difference between analytical and numerical method for ΔT_c and Δp and is similar to liquid water results [Table 9]. That clearly sets a direct relation between Reynold number and the validity of the analytical model. So, as Dr. Guan stated in his thesis (1), the Dittus-Boelter expression works as from a certain value of Reynolds. He wrote it was from 10^4 [Equation 16], which tallies with the results just obtained in Table 17. Analogously, the same happens with the pressure drop and [Equation 18].

Figure 22 represents the visual comparison for these five new cases.

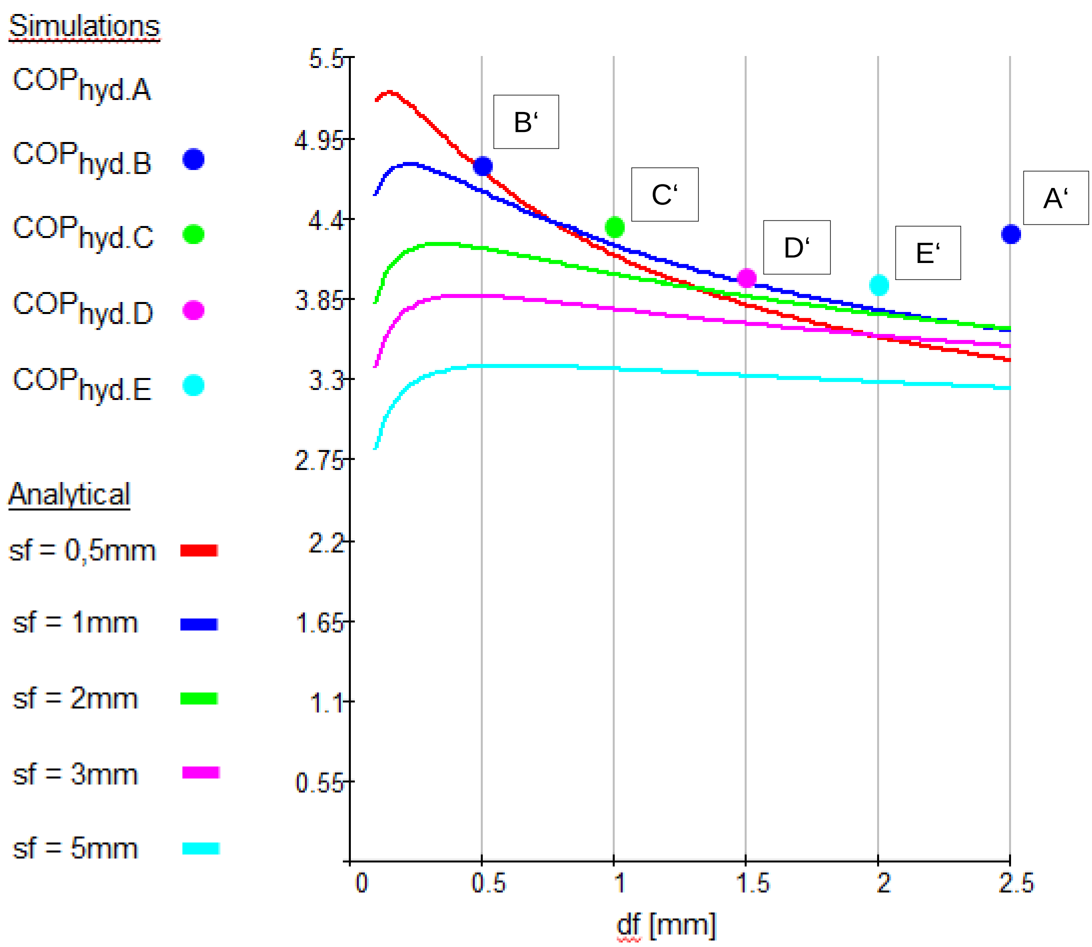


Figure 22: Graphical comparison between analytical and numerical results (water-glycol with P_{hyd}=600W)

It can be noticed the difference between Figure 22 and Figure 21. The points that represent the simulation results are close to their respective curves. The whole distribution of the graphic is really similar to Figure 20 (liquid water). With similar Reynolds numbers there are similar distribution when the analytical and numerical comparison is plotted into the same graph.

Obviously, in reference to the vertical axis it is not acceptable to have those low values of the COP_{hyd}. These are due to the high hydraulic power that has been imposed with the only aim of demonstrating that with high Reynolds numbers Dittus-Boelter´s equation is accurate for predicting heat transfer behaviour in applications like the one here studied.

In conclusion, for making possible the use of Dittus-Boelter equation for the analytical calculations of plate-fin heat sinks with water-glycol as a fluid coolant it would be necessary to increase its Reynold number to values around 10^5. In terms of kinetics it would mean increasing the value of the hydraulic power, what it can be impossible in most of Power Electronics applications.

The other option that remains is trying to modify Dittus-Boelter equation so that it suits applications with low Reynold numbers. This has been a common activity in the history of heat transfer studies. Many new correlations that suit different applications have been obtained modifying a previous one and analysing the results.

Finally, before ending with the water-glycol results, it has to be said that few other calculations with water glycol were made that here have not been represented. They were done for trying to find a solution that could predict accurately the behaviour of water-glycol´s heat dissipation. They did not provide the sought accuracy, although they were useful to discard different possibilities that could have been right at the beginning and therefore validate in a stronger way the previous conclusion around the validity of using Dittus-Boelter equation for larger Reynold numbers. The following table shows directly the comparison between simulation and analytical methods of the mentioned calculations.

		Comparison of average differences	
Coolant	**Set-up**	**ΔT_c [%]**	**Δp[%]**
Water-glycol	ANSYS - Laminar model	**54,74%**	**28,20%**
	ANSYS - Transition model	**49,12%**	**36,36%**
	Gnielinski correlation for Nu	**72,45%**	**47,41%**

Table 18: Extra calculations for water-glycol

The three calculations that are summarized in Table 18 represent three different model set-ups that were applied to cases A, B, C, D, and E [Table 6]. The first line, *ANSYS – Laminar model,* makes reference to a change of viscous model that was made in ANSYS simulations. As it is said in 5.2.3, the simulations are done with the *k-omega* model for an optimal prediction of turbulence. However, as the Reynolds number was so low for water-glycol, the *Laminar* model was used for discarding the possibility of the ANSYS set-up being wrong. For the same reason was the ANSYS transition model tried. As we can see in the percentage comparison, both models did not improve the difference, which it is even higher in terms of ΔT_c. That means the ANSYS viscous model was not the problem, it remarks that it was the analytical correlation used.

As a last testing, the Gnielinski expression for Nusselt correlation was introduced in the analytical model. Dr. Guan had already explained in his thesis that it could be a possible correlation for obtaining accurate models in plate fin heat sinks in even lower Reynolds:

$$Nu=\frac{\left(\frac{f}{8}\right)\cdot(Re-1000)\cdot Pr_f}{1+12{,}7\cdot\sqrt{\frac{f}{8}}\cdot\left(Pr_f^{\frac{2}{3}}-1\right)}\cdot\left(\frac{Pr_f}{Pr_w}\right)^{0{,}11} \qquad 2300<Re<10^6$$

[Equation 27]

Substituting the correlation in [Equation 27] for Dittus-Boelter expression, the results do not improve. They even get way worse. That means Gnielinski equation is also not valid for predicting heat transfer in plate fin heat sinks with the actual working conditions.

These three last comparisons ratify even more the conclusions stated after the analysis of water-glycol shown in Table 15, Table 16 and Table 17. With water-glycol as fluid coolant, it would be necessary then to investigate in new Nusselt correlations for low Reynold numbers or if the real-life application permits it increase the Reynold numbers to values higher than 10^4 and around 10^5.

6.3 Calculation with dielectric fluid coolants

Parallel to water-glycol calculations, the three dielectric coolants which were introduced in chapter 3 were studied. These was an important point because the main objective of this thesis [2.1] was to analyse if Galden HT-135, Novec 7500 and Fluorinert FC-3283 could be used as fluid coolants with the same analytical model as the one proved valid for liquid water. It was also interesting to know how the fluids worked respect to liquid water, that is if they could manage to dissipate heat in a way that they could be considered as good substitutes of water.

Table 19 shows the value of the variables for the three dielectric coolants distributed in each of the five cases.

		Value				
Coolant	**Concept**	**Case A**	**Case B**	**Case C**	**Case D**	**Case E**
Galden HT-135	P_{hyd} [W]	0.68	0.68	0.68	0.68	0.68
Novec 7500	P_{hyd} [W]	0,55	0,55	0,55	0,55	0,55
Fluorinert FC-3283	P_{hyd} [W]	0,60	0,60	0,60	0,60	0,60
Common variables	d_f [mm]	2,50	0,50	1,00	1,50	2,00
	s_f [mm]	1,00	1,00	2,00	3,00	5,00
	u_c [m/s]	1,91	1,37	1,81	2,11	2,47

Table 19: Value of the variables in the different cases with the dielectric fluids as coolants.

Repeating the process once more, d_f, s_f and u_c are the same as defined for liquid water [Table 6] and the parameter that changes for each coolant is the hydraulic power, what is due to fulfil the target of keeping the velocities the same for each case respectively. As It can be seen, the whole three values of P_{hyd} are representative of what it could be found in a real application for Power Electronics.

Table 20 and Table 21 show the analytical and numerical results respectively.

Analytical results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**Δp [mbar]**
Galden HT-135	A	14040,00	85,00	85,90	**0,90**	**20,26**
	B	2633,00	85,00	88,98	**3,98**	**61,62**
	C	6479,00	85,00	87,23	**2,23**	**46,52**
	D	11480,00	85,00	86,54	**1,54**	**39,89**
	E	15460,00	85,00	86,19	**1,19**	**39,78**
Novec 7500	A	18320,00	85,00	85,87	**0,87**	**16,70**
	B	3437,00	85,00	88,84	**3,84**	**49,90**
	C	8458,00	85,00	87,15	**2,15**	**37,66**
	D	13790,00	85,00	86,49	**1,49**	**32,30**
	E	20190,00	85,00	86,15	**1,15**	**32,21**
Fluorinert FC-3283	A	20420,00	85,00	85,89	**0,89**	**18,23**
	B	3829,00	85,00	88,89	**3,89**	**54,46**
	C	9423,00	85,00	87,20	**2,20**	**41,11**
	D	15370,00	85,00	86,53	**1,53**	**35,23**
	E	22490,00	85,00	86,19	**1,19**	**35,15**

Table 20: : Results of the analytical calculation with the dielectric fluids as coolants.

Simulation results						
Coolant	**Case**	**Re []**	**T_{c0} [°C]**	**T_{cf} [°C]**	**ΔT_c [°C]**	**Δp [mbar]**
Galden HT-135	A	14040,00	85,00	86,12	**1,12**	**23,16**
	B	2633,00	85,00	90,78	**5,78**	**91,61**
	C	6479,00	85,00	87,90	**2,90**	**60,15**
	D	11480,00	85,00	87,06	**2,06**	**52,40**
	E	15460,00	85,00	86,59	**1,59**	**42,40**
Novec 7500	A	18320,00	85,00	86,09	**1,09**	**20,48**
	B	3437,00	85,00	90,14	**5,14**	**72,38**
	C	8458,00	85,00	87,76	**2,76**	**47,27**
	D	13790,00	85,00	86,99	**1,99**	**41,41**
	E	20190,00	85,00	86,55	**1,55**	**33,98**
Fluorinert FC-3283	A	20420,00	85,00	86,10	**1,10**	**22,16**
	B	3829,00	85,00	90,11	**5,11**	**77,78**
	C	9423,00	85,00	87,80	**2,80**	**50,75**
	D	15370,00	85,00	87,04	**2,04**	**44,84**
	E	22490,00	85,00	86,60	**1,60**	**36,99**

Table 21: Results of the simulation with the dielectric fluids as coolants.

Table 20 and Table 21 show the results distributed by type of coolant and then by each case. Observing the Reynold number column, it can be noticed that the values are way higher than compared to water-glycol [Table 11]. Moreover, they are in a similar order as liquid water [Table 7], although with a lower value. That happens because of the higher kinematic viscosity that the three dielectric fluid coolants have when compared to water [Table 3]. Despite this, watching the Reynolds number it can be anticipated that the analytical model will function better with the three dielectric fluids than with water-glycol.

Observing the column of the temperature differential (ΔT_c), it can be seen that the fluid which has a higher value is Galden HT-135. This occurs in both analytical and numerical results and this uniformity in the values can be understood as a good sign before approaching the comparison. Returning to Galden HT-135, its higher values of ΔT_c mean that the thermal power will also be higher; therefore, the heat dissipation will be more favourable. Fluorinert FC-3283 is the second with higher values of temperature differential and finally comes Novec 7500.

Finally and before analysing the comparison between simulations and analytical procedure, if the last column (Δp) of Table 20 and Table 21 is observed, it can be realised that the pressure drop behaves in the same way as the temperature increasing. In this aspect, Novec 7500 would be the one with less pressure drop followed by Fluorinert FC-3283 and finally would come Galden HT-135.

The last two paragraphs show the importance of having cleared which way of optimisation is wanted for a cooling system based in plate-fin heat sink technology. It will be always a matter of arriving to the equilibrium wanted between the parameters that influence the most in the design of the heat sink for the application for which has to be used (heat dissipation, pressure drop, material volume or volumetric flow).

Firstly, though, the analytical model should be validated. In order to do the comparison, Table 22 was deigned analogously to Table 20 and Table 21, so that the difference between analytical results and ANSYS results are distributed by fluid coolant and then by each different case.

		Analytical		Simulation		Difference [%]	
Coolant	**Case**	**ΔTc [ºC]**	**Δp [mbar]**	**ΔTc [ºC]**	**Δp [mbar]**	**ΔT_c**	**Δp**
Galden HT-135	A	0,90	20,26	1,12	23,16	**20,00%**	**12,51%**
	B	3,98	61,62	5,78	91,61	**31,04%**	**32,73%**
	C	2,23	46,52	2,90	60,15	**23,30%**	**22,66%**
	D	1,54	39,89	2,06	52,40	**25,42%**	**23,86%**
	E	1,19	39,78	1,59	42,40	**25,55%**	**6,18%**
Novec 7500	A	0,87	16,70	1,09	20,48	**20,55%**	**18,46%**
	B	3,84	49,90	5,14	72,38	**25,38%**	**31,06%**
	C	2,15	37,66	2,76	47,27	**22,32%**	**20,32%**
	D	1,49	32,30	1,99	41,41	**25,14%**	**21,99%**
	E	1,15	32,21	1,55	33,98	**26,02%**	**5,22%**
Fluorinert FC-3283	A	0,89	18,23	1,10	22,16	**19,34%**	**17,76%**
	B	3,89	54,46	5,11	77,78	**24,03%**	**29,98%**
	C	2,20	41,11	2,80	50,75	**21,48%**	**18,99%**
	D	1,53	35,23	2,04	44,84	**24,79%**	**21,44%**
	E	1,19	35,15	1,60	36,99	**26,00%**	**4,96%**

Average Galden	**25,06%**	**19,59%**
Average Novec	**23,88%**	**19,41%**
Average Fluorinert	**23,13%**	**18,63%**

Table 22: Comparison between analytical and simulation methods for dielectric fluids as coolants. Difference calculated respect to the ANSYS values.

Observing Table 22, it can be quickly seen that the average values are quite close. For temperature differential they are between 23% and 25% and for pressure it is even smaller (18% to 19,50%). It is important to have lower average difference than the one obtained for water-glycol so that it can be studied the validity of the analytical method. The dielectric fluids, which all have values in close intervals; although they do not arrive to the Reynolds numbers values as liquid water does they are in the same order and quite close. This is reflected in the lower average difference when it is compared to water-glycol.

As it can be also seen in Table 22, Galden is the coolant that presents a bigger difference in the average concerning temperature. It is also the one that has a bigger divergence in pressure drop, although in this case it is too small to be considered. However, it is important to remark it because Galden is also the dielectric fluid coolant that has the lower Reynolds number values [Table 20]. Even observing the other two dielectric coolants, FC-3283 and Novec 7500, it can be noticed that in the same way the one with higher Reynold numbers (FC-3283) is also the one with lower average difference. Even with such a tiny difference the dependence on

Reynolds number is remarkable. It can be here concluded that the Reynolds number is crucial in order to try to solve the heat transfer issue with an analytical method.

Figure 23 shows graphically the COP_{hyd} comparison between numerical and analytical methods for the three fluid coolants. Each coolant results are represented in their respective graph and they all share the same legend.

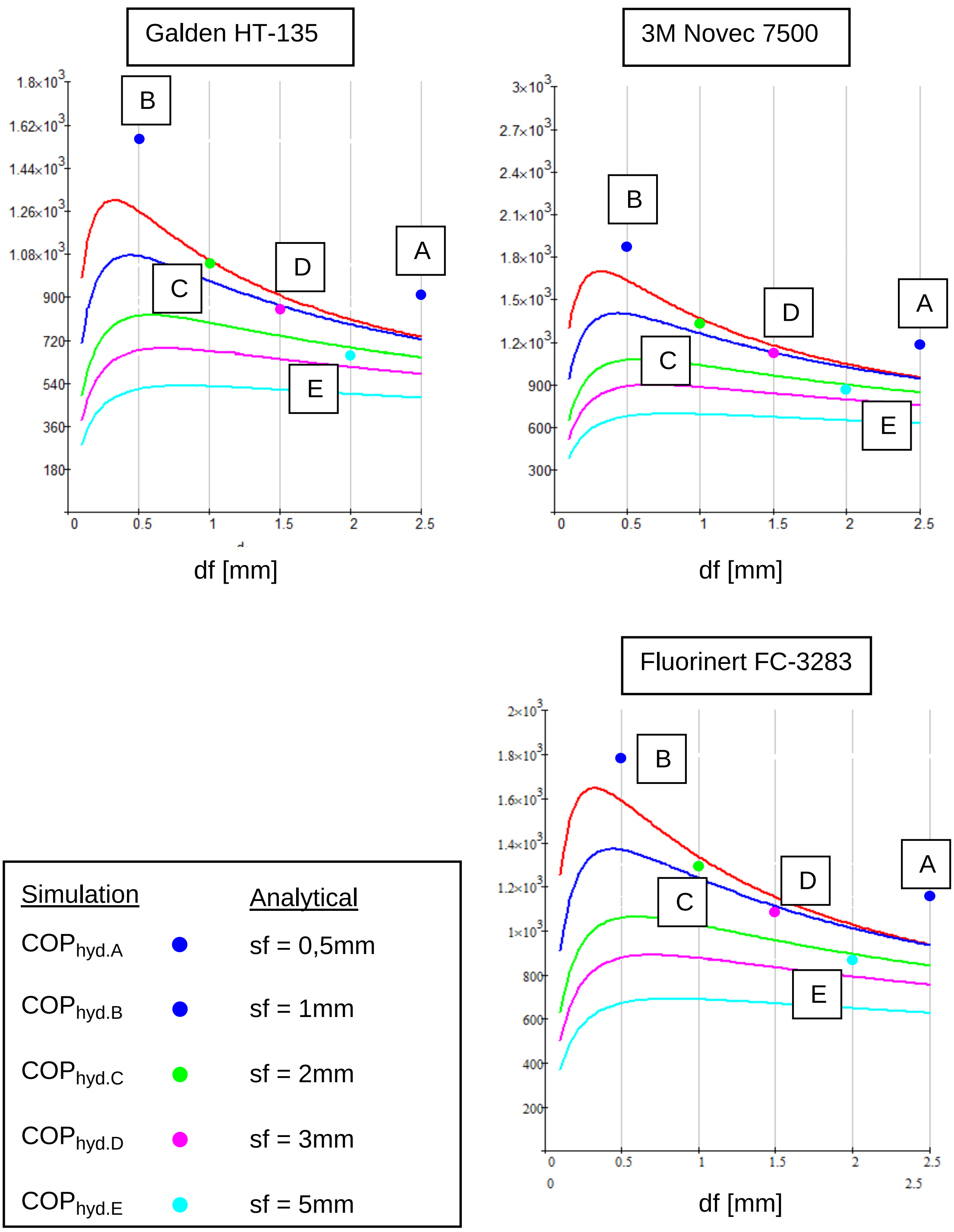


Figure 23: Graphical comparison between analytical and numerical results for the dielectric coolants.

Observing Figure 23, the first thing that can be realized is that the graphical distributions are close to being identical. The distribution of the points respect their respective curves is the same in the three graphs. This fact continues with the confirmation of coherence and uniformity in the results that was stated both in the liquid water results and in the water-glycol ones. For the three different dielectric coolants ANSYS results are always above the analytical ones and they are spread in the plot on a way that has always the same shape. This can be a determinant feature at the moment of deciding to accept or refute the analytical model as later it is going to be explained.

Comparing the three graphs between them in terms of heat dissipation efficiency, that is by means of their COP_{hyd}. Analysing the three plots and looking to their respective vertical axes it can be extracted that Novec 7500 is the coolant with higher values of COP_{hyd} both in analytical and simulation models. FC-3283 has close values to Novec and this makes a very small operative difference. With Galden differences are way bigger, being this one the worse of the three in terms of thermal efficiency. These results have a coincidence with the values of hydraulic power (P_{hyd}) [Table 19]. So, as it can be logically concluded from the expression in Equation 26, a higher value for P_{hyd} leads to a decrease of the thermal efficiency referred to it. This topic is deeper treated on chapter 7.

In reference to the comparison in Figure 23 of each graph by itself it is important to notice that the vertical distance between ANSYS´ points and the analytical curves is closer in terms of size to the one that can be found in Figure 20 (liquid water results) than to the one that can be found in Figure 21 (first water-glycol results obtained). This is also a relevant fact to consider because it confirms that the dielectric coolants with the same values of the variables as the ones as liquid water, can have their thermal behaviour predicted in a way that is close to water. In this comparison it can also be observed that liquid water efficiency values are way higher than the three dielectric fluids, approximately by 4000 – 5000 units of COP_{hyd}, as it can be deducted by the better thermal behaviour (more heat transferred) and the lesser hydraulic power needed to move the fluid through the circuit.

Nevertheless, the objective is not to find a dielectric fluid coolant that can be as good as water for heat transfer applications but finding one (or more) that can be a worthy substitute to liquid water in applications where this last one cannot be used because of physic matters (like the one that concerns this thesis).

The following paragraphs intend to do a summary of what has been done in this chapter and show the conclusions that were extracted after all the results obtained and comparisons made.

6.4 Conclusions on the results obtained

During this chapter the whole results obtained after applying the processes described in chapters 4 and 5 were chronologically showed and analysed. Firstly, liquid water results proved to have accuracy even better than expected when compared simulation and analytical models [Table 9 and Figure 20]. This successful analysis led to advance to the next step which involved the calculations with water-glycol as a fluid coolant. Nevertheless, these second results were not as successful as the first ones. However, they allowed the testing with different conditions [Table 14 and Table 18] which proved the importance of Reynolds numbers when it comes to thermohydraulics cooling predictions with analytical models. This feature of the Reynolds number was reaffirmed during the step after, where the three different fluid coolants were tested and analysed. Referred to these three coolants (Galden HT-135, Novec 7500 and Fluorinert FC-3283), the analysis of their results led to favourable conclusions in terms of the analytical model, because of the accuracy in the numerical-analytical comparison [Figure 23 and Table 22].

The conclusions on the validity of the analytical models after the analysis of the results are summarized in each of the next three sub-chapters.

6.4.1 Liquid water

Liquid water is the easiest fluid to analyse because it was the one that proved to have the closest difference between numerical and analytical models. An average of 9,21% of percentage difference respect to simulation results is a value that totally confirms the validity of Dittus-Boelter equation for calculations in the Reynolds numbers conditions that appear at Table 9. It can also be said that it was expected a value between 10-15%, not even less than 10% as it was obtained. Lesser than that it could begin to seem unreal because the margin of error that has to be always considered between two different ways of obtaining the results.

In terms of the pressure drop, a difference of 14,77% was obtained which is a bit bigger than the heat transfer one. Nevertheless, it is considered to be inside the interval formed by the logical values that can be expected after considering the margin of error mentioned before. Therefore, the analytical method for liquid water works both for heat transfer (Dittus-Boelter) and for pressure drops (Culham).

6.4.2 Water-glycol

Water-glycol was the fluid which had the most number of calculations and analysis. That is because it also had the worst accuracy in the first results obtained, so a few more tests were done with the aim of achieving either satisfying results or the explanation for such a big difference.

However, referred to the percentage differences obtained in the first calculations (40,42% for ΔT_c and 44,26% for Δp) it is thought that they are too high for validating the analytical model directly. That is why water-glycol is not included in chapter 7 as one of the fluids studied for obtaining their operating curves.

The conclusion at which is arrived to when referring to water-glycol is that to use actual analytical model its Reynolds numbers should be increased substantially. Nevertheless, this would mean to increase the hydraulic power to values that are totally inadequate for fluid cooling in Power Electronic applications (only for getting close values to liquid water Reynolds numbers 600W of P_{hyd} were needed which is already inadmissible). So, the final conclusion is that for obtaining a valid analytical model to be used in the actual conditions of constants and variables, a modification on the Nusselt correlation and on the friction factor correlation should be done. A new correlation that suited the low Reynolds of current water-glycol analysis would be the correct way of getting a valid analytical model.

6.4.3 Dielectric fluid coolants

In regards of the dielectric fluid coolants (Galden HT-135, Novec 7500 and FC-3283), the average differences calculated are in between 23% and 25% for ΔT_c and between 18% and 20% for Δp. These results show a difference between numerical and analytical models a bit too high than the one it is desired to achieve (10%-15%). Analogously, the Reynolds numbers of the cases studied with these coolants are all beneath liquid water ones so that is the reason for a bigger difference in the comparison.

Although it would be also interesting to apply a modification to the Nusselt correlation and to the friction factor it was thought that the percentage differences values are so close to the desired ones that it can be taken into consideration a study with the current analytical model. It was also crucial in this decision the fact that in all the graphs that are represented in Figure 23 the distribution of the simulation points superimposed to the analytical curves follow a same pattern. This coherence and uniformity in the results together with the relatively low numerical-analytical difference allows to consider as valid an analysis of the operation of the mentioned three dielectric fluid coolants with the current analytical model.

The next chapter shows different plots for liquid water and the three dielectric fluid coolants with the objective of making a comparison between them and extracting conclusions referred to their usefulness within heat dissipation in Power Electronics applications.

7 Diagrams for plate-fin heat sink designs

This last chapter was included with the objetive of showing the way of operation of the different fluid coolants subjected to different conditions. In the following four pages are plotted a total of four figures [Figure 25, Figure 26, Figure 27 and Figure 28] and each one of them include eight diagrams for a total of thirty-two graphs. Each diagram represents the analytical model curves under determined conditions. On the left side of the page there is plotted both the thermal efficiency respect to the hydraulic power (COP_{hyd}) and the volumetric flow, and on the right side there is the pressure drop. The pages are distributed by the hydraulic power value (P_{hyd}), hence the first page with graphs correspond to 0,05W, the second to 0,1W the third to 0,2W and the last one to 0,5W. Each page is also distributed in the four fluid coolants shown: 3M Novec 7500, Galden HT-135, 3M Fluorinert FC-3283 and liquid water.

As it has been said, each sheet corresponds to a determined value of P_{hyd}. Therefore, the study resides in the different operative curves of the analytical method when changes in hydraulic power is applied to the cooling circuit. This can be interesting for the reader in order to know how the thermal and mechanical behaviour of the fluid coolants change when they are subjected to different hydraulic powers. From the point of view of the design of the cooling circuit it is important to decide the value of power used to keep it running. The selected values were four and they belong to an interval which is representative for Power Electronics real applications [0,05W , 0,5W] (1). With it, the reader can have an idea of how its cooling system would work depending on the hydraulic power available.

As a last comment for this introduction, it is necessary to say that in each diagram five curves are represented, each one for a different fin thickness. The horizontal axis always represents the separation between fins and the Y-axis represents the COP_{hyd}, the V_{flow} and the Δp respectively. That combined with the changes in the P_{hyd} allows the achievement of an overall understanding of how the geometry and power affect the behaviour of the plate-fin cooling systems for Power Electronics.

Figure 24 shows the legend that allow to identify each curve with its colour. In the following four sheets the diagrams are represented and afterwards there is a brief explanation on them.

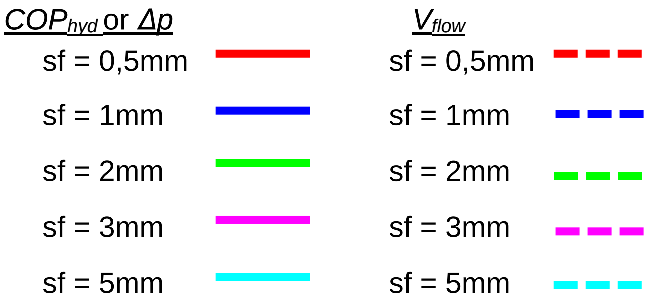


Figure 24: Legend information for Figures 25, 26, 27 and 28.

P_{hyd} = 0,05W

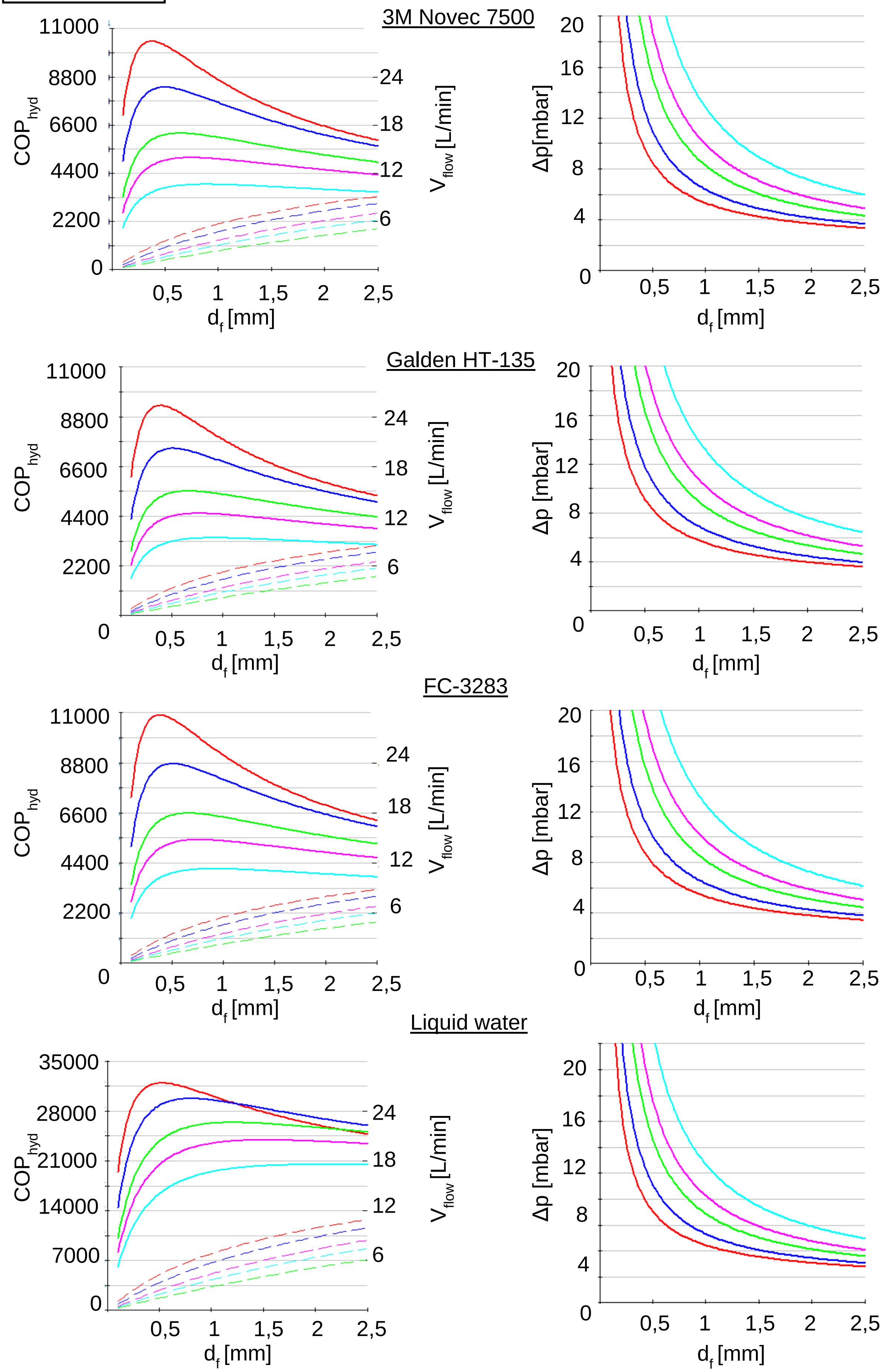


Figure 25: COP_{hyd} and pressure drop graphs for the three fluid coolants and water at P_{hyd} = 0,05W.

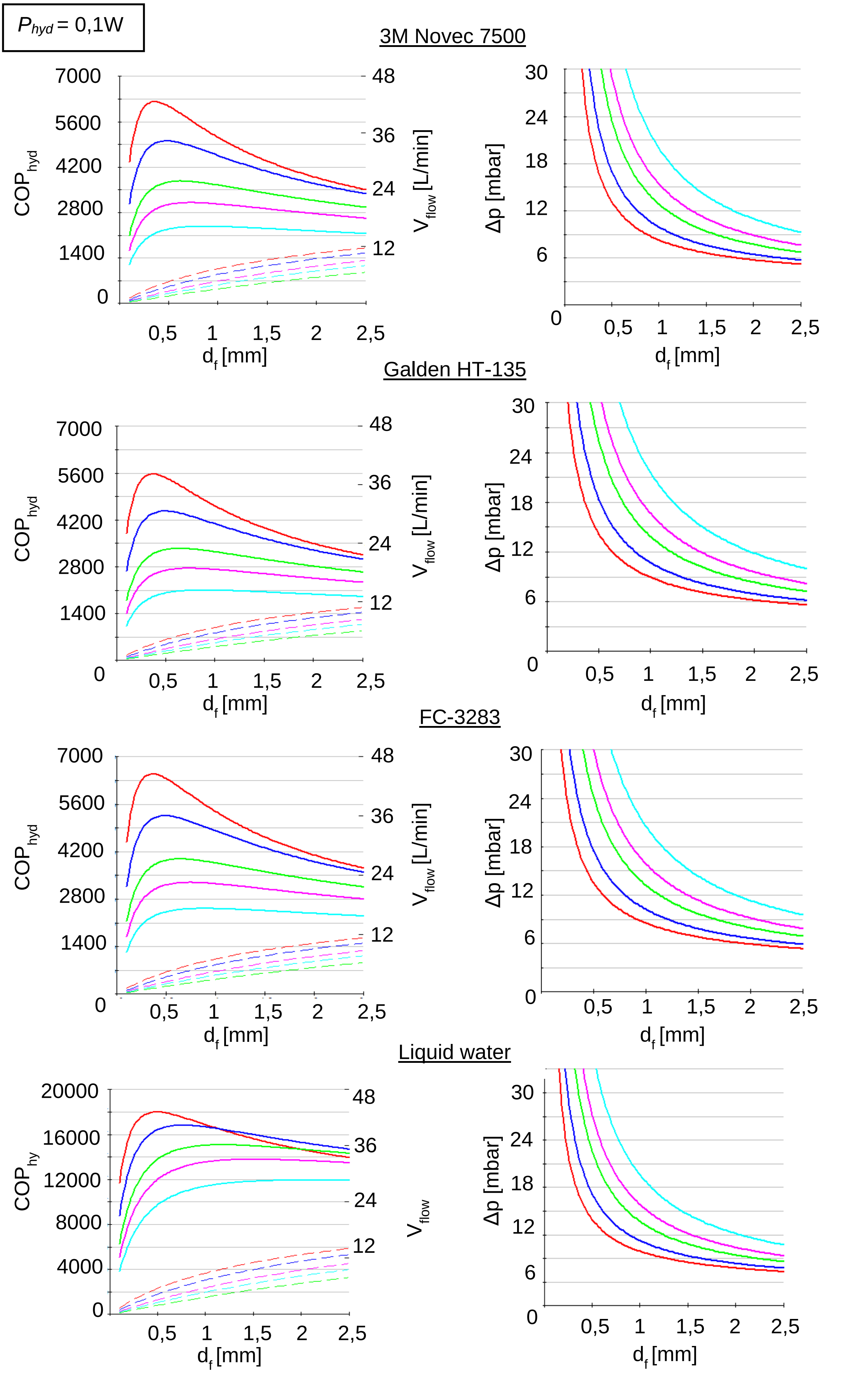
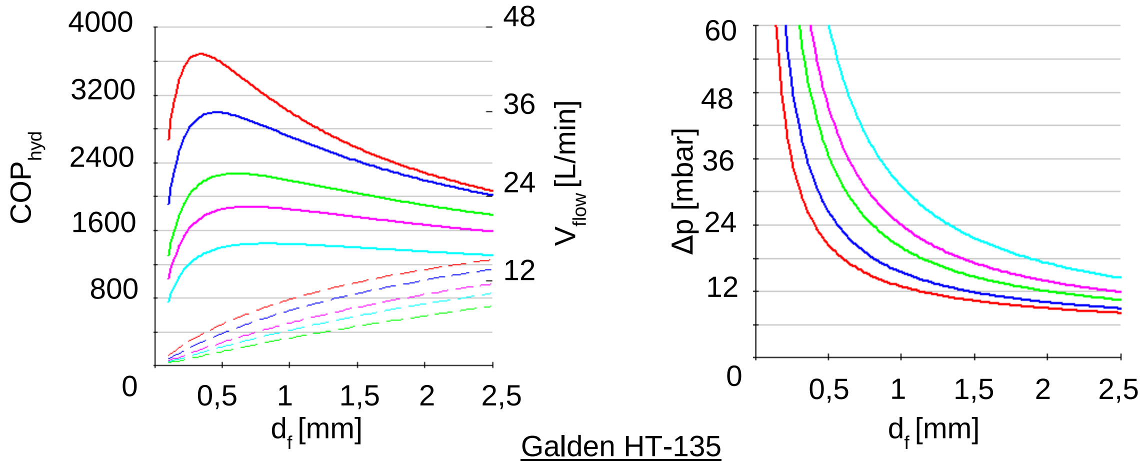


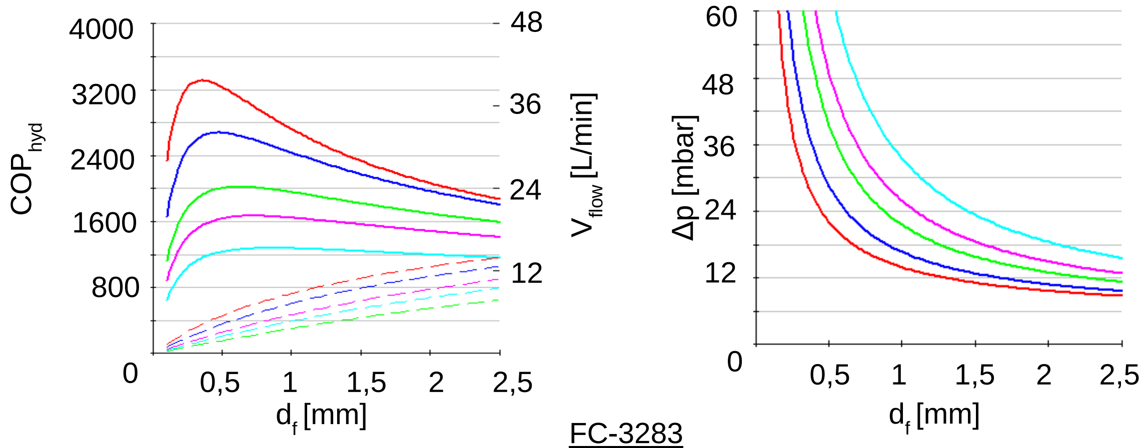
Figure 26: COP_{hyd} and pressure drop graphs for the three fluid coolants and water at P_{hyd} = 0,1W

$P_{hyd} = 0{,}2W$

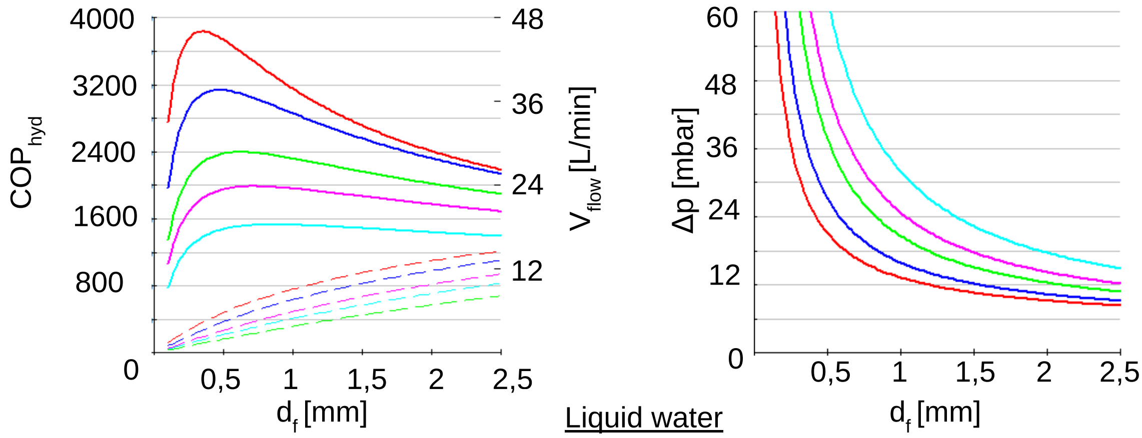
3M Novec 7500



Galden HT-135



FC-3283



Liquid water

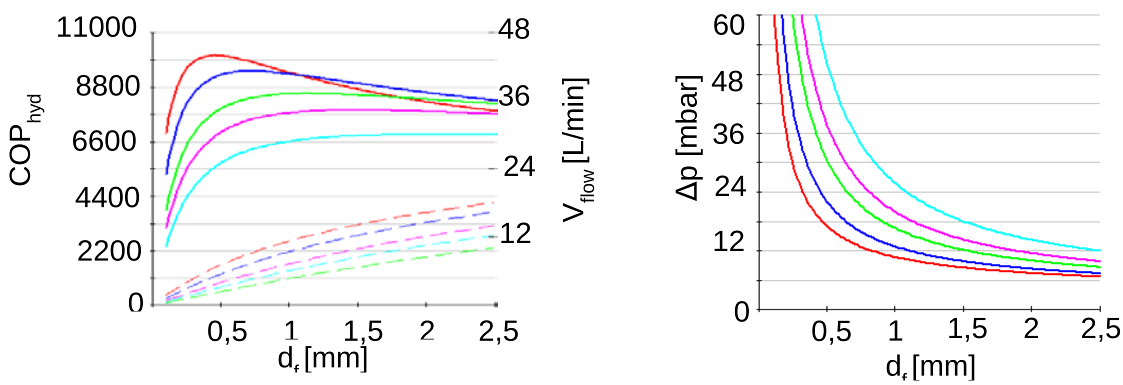
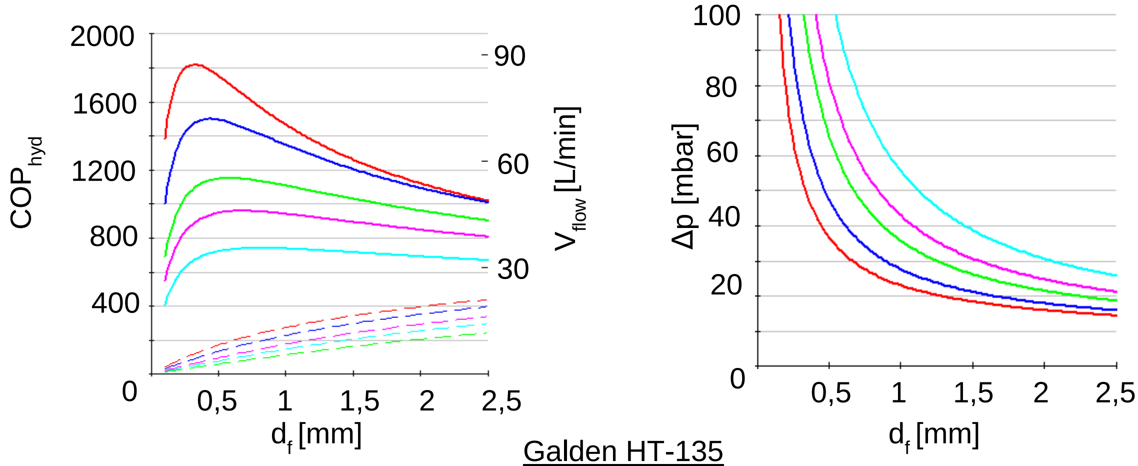


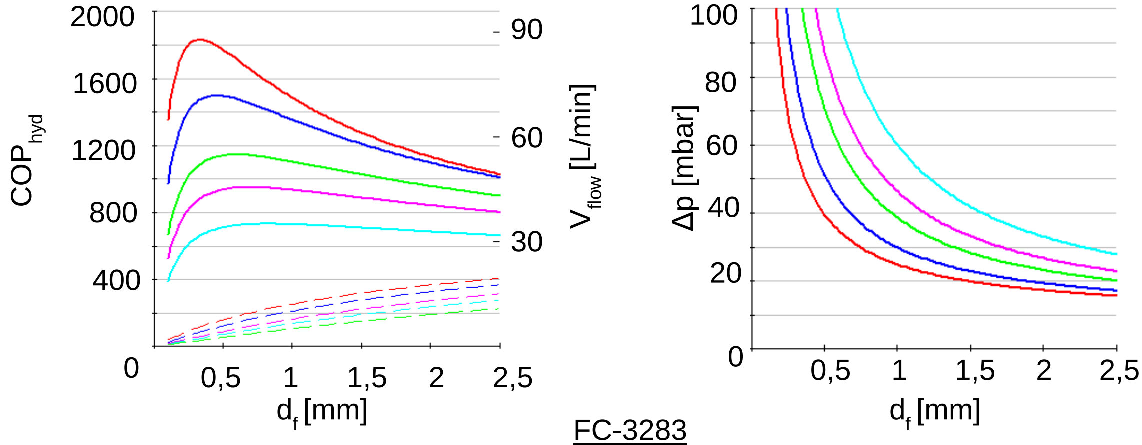
Figure 27: COP_{hyd} and pressure drop graphs for the three fluid coolants and water at $P_{hyd} = 0{,}2W$

P_{hyd} = 0,5W

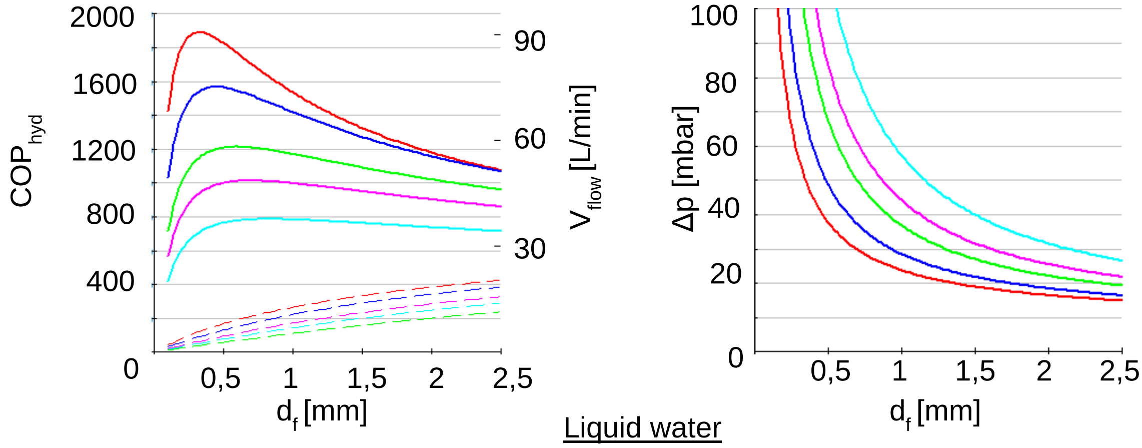
3M Novec 7500



Galden HT-135



FC-3283



Liquid water

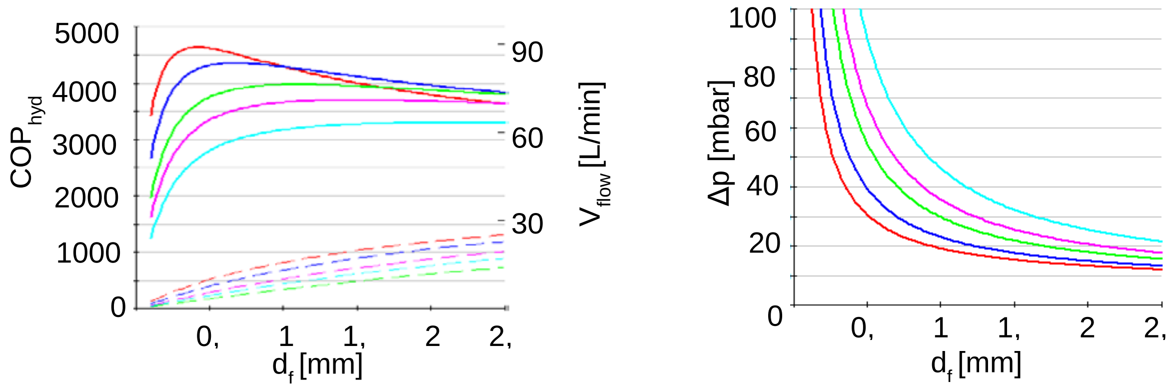


Figure 28: COP_{hyd} and pressure drop graphs for the three fluid coolants and water at P_{hyd} = 1 W

For the interpretation of the amount of diagrams that can be seen in the last four pages, firstly the heat dissipation is going to be analysed. For that it is needed to focus on the COP_{hyd} diagrams, that is on the left side of each page. At first only the three dielectric coolants are going to be analysed and later compared to liquid water. Just comparing the scale of COP_{hyd}'s axis between Figure 25, Figure 26, Figure 27 and Figure 28 it can be seen that the increase of hydraulic power is inversely proportional to the thermal efficiency, which decreases from the lowest value of P_{hyd} to the highest. It is a fact that turbulence improves heat transfer (8), therefore with a bigger hydraulic power, thus bigger volumetric, velocity and turbulence the thermal power increases. In fact, if the conversion of COP_{hyd} to P_{th} is done with Equation 26, the thermal power results to increase every time hydraulic power does so. It is interesting then, to know that the thermal efficiency does not do so and with less hydraulic is achieved a better value of $COP_{hyd.}$ Therefore, systems with a lesser value of P_{hyd} will always be more efficient in terms of the energy waste.

Continuing with the thermal analysis and observing closely the dielectric fluid coolants COP_{hyd} diagrams in Figure 25, it can be seen that the coolant that the fluid coolant that has the better results for thermal efficiency and also for heat dissipation (as it is a comparison with fixed hydraulic power of 0,05W) is FC-3283, whose curves are situated the highest of the three dielectric fluids. It is followed by Novec 7500 and at last comes Galden HT-135. The curves have the same shape for each fluid, which allows to easily compare their values. In Figure 25 the difference between the peak value (red curve) of FC-3283 and Novec is about 500 unities of COP_{hyd} and between FC-3283 and Galden is about 1500 unities. This difference gets reduced as the curve advances to the left because of a bigger inclination of the FC-3283 curve, but it remains always with clear values above the other two coolants. The difference continues the decrease proportionally as the hydraulic power increases although always FC-3283 is the coolant with the best efficiency. If Figure 28 is observed, it can be seen that Novec and Galden's diagrams are practically identical but taking a closer look on their respective red curve, it can be seen that Galden's one is a bit above Novec's one. That means that the for higher hydraulic power values Galden can procure a better heat dissipation.

As a conclusion for last paragraph it can be stated that for low values of hydraulic power, there exists a clear *winner*. This would be FC-3283, followed by Novec 7500 and in third place Galden HT-135. Nevertheless, when the hydraulic power increases, the difference gets lesser and the fluids work in a nearer operative values.

For ending the thermal comparison, the effect of geometry must be compared. In each diagram the optimum is obtained with a distance between fins d_f of 0,4 to 0,5 and with a fin thickness of 0,5 (the least of all of them). Small fin thickness and small distance between fins combined just means high number of fins. Therefore, it is deducted from this statement that high numbers of fins yield to better heat transfer. The theory of fins says just that, that they are made to enlarge the area of heat transfer, so it is a coherent result that higher number of fins is related with more thermal power transmitted. Smaller d_f like 0,1mm or 0,2mm is considered to be too small for even the flow to circulate in a proper way, thus to have a proper heat transfer. That is why at the left side of the diagram the values for COP_{hyd} decrease in a severe way.

Consequently to what is said in the last paragraph, as the d_f increase and the s_f does the same, the heat dissipated also decreases. Nevertheless, it is seen in each COP_{hyd} diagram that the bigger is s_f value the faster it decreases thermal efficiency. Hence that the red or blue curves have a more pronounced inclination than the pink and cyan ones. It can be made the deduction that if the distance between thins (d_f) continued to increase to values higher than 2,5mm the red line would star to cross the other ones and also would do the blue one. This can be seen accurately in each figure for the case of liquid water, where at the right side of the diagrams the red curve has already lower values than the blue, the green and the cyan ones.

Comparing the three dielectric coolants to liquid water, it is seen that liquid water has way higher values than the three dielectric coolants. This is considered normal as water is the best heat transfer fluid for these kind of temperatures available in the market. However, the aim of the diagrams is to show the operating points of the three newly introduced dielectric coolants as they can be a substitute of water in applications were this last one cannot be used.

Finally, it is time to do a brief interpretation of the pressure drop diagrams (Δp), which are situated at the right side of Figure 25, Figure 26, Figure 27 and Figure 28. Three facts can be highlighted. The first one is that as the hydraulic pressure increases, so it does the pressure drop. Coherently to that last statement, as the hydraulic power is kept stable and the fin thickness (s_f) increases, therefore the volumetric flow decreases, the pressure drop also increases. This is resumed in [Equation 23]. However, a parameter that affects the most in the moment of designing because of the pressure drop is the distance between fins (d_f). As the space gets thinner the pressure drop increases exponentially. It is important to be aware of that in order to design a correct flow through the cooling circuit.

The choice of geometry of the plate-fin heat sink is always in hands of the engineer. It is important that other external factors that here are not included can be essential for choosing one or another configuration. For example, in some industry applications such low values of distance between fins as 0,5mm or 1,0mm are not allowed because particles that circulate through the circuit can block the heat sinks. In the end is a decision that has to be made with all the variables of the determined application once analysed. This chapter just tries to make a first approach to this decision and can be useful for a reader to have a first idea on which configurations could be useful to its application and which not.

8 Conclusions of the thesis

In this thesis an analysis of heat dissipation in Power Electronics by means of plate-fin cooling systems has been done. Overheating of the electronic components in Power Electronics systems is always an issue and nowadays is increasing as the applications that use the DC-DC technology are growing in quantity from year to year. That is why new researches in the way the cooling is done have to be made. This paper offers the reader a view of the analysis of five fluids (liquid water, water-glycol, 3M Novec 7500, Galden HT-135 and Fluorinert FC-3283) that could possible offer solutions for real life applications.

The first objective was to validate the analytical model with the numerical model for liquid water. This was completed successfully in chapter 6.1. Once obtained this satisfactory results for liquid water, the other fluids were tested. Referred to water-glycol, the results were not as satisfactory as expected so a few more calculations with different configurations were made. That led to the proof of the crucial importance that Reynolds number had in the validation of the analytical correlations. Figure 29 shows the percentage difference between analytical and simulation methods vs. the Reynolds number for liquid water, water-glycol, Galden HT-135 and FC-3283. The orange line indicates the tendency of the diagram and it is clearly seen that, except for some singular cases, as the Reynolds number gets bigger, the difference between simulations and analytical correlations gets smaller.

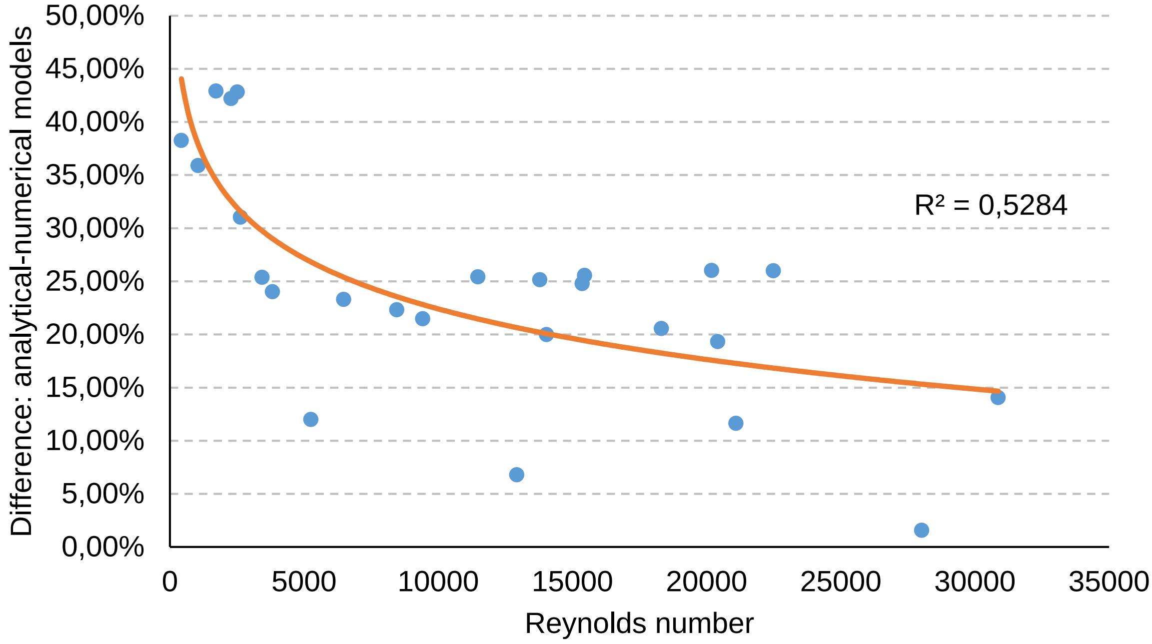


Figure 29: Reynolds number correlations related to the difference between analytical and numerical models.

According to the line of tendency of the logarithmic curve that is drawn in Figure 29, which was the one with higher correlation, as the Reynolds number grows the percentage difference between the analytical and numerical models decreases and it can be seen that it gets stable between 10% and 15% of difference, where liquid water´s difference is. The points below 10% are considered exceptions that can always appear. A minimum margin of error has to be always considered between numerical and analytical results, that is why the interval between 10% and 15% is the one that is searched. In conclusion, with higher Reynolds number than 5000 unities the percentage difference is reduced to 25% or less which it is interesting in order to start to consider the analytical model as valid.

The analysis of the three dielectric coolants went better than the water-glycol one and the values obtained in the results were close to the intervals of acceptance searched. Because of the coherence and uniformity if their results, both Dittus-Boelter correlation for Nusselt number and Culham expression for the friction factor were considered as valid for doing an analysis of their operation curves.

Finally, a set of operation curves for the three dielectric fluids and for liquid water were drawn in order to give the reader an idea of how the mentioned fluids work subjected to different configurations in terms of geometry and hydraulic power. Moreover, each one of these different configurations is possible in a Power Electronics application for industry.

With that said, it can be concluded that the initial objectives of the thesis were fulfilled. The analytical and the simulation methods were compared and analysed and conclusions were extracted from their analysis. With the fluid that had non-acceptable results (water-glycol) further research was made to reach the reason why the analytical model did not give the results as expected. On the other hand, with fluids that had results considered as valid (liquid water and dielectric fluids) their operation curves were drawn and analysed.

Finally, it is important to mention that from here different paths can be followed to continue with the investigation regarding heat sink cooling in Power Electronics. The first of them is the research on new Nusselt correlations that can predict more accurately the values of heat transfer and pressure drop in low Reynolds number for plate-fin heat sinks. On the other hand, the analysis of the application of the fluid coolants in the industry for the usage that in this paper is described should be done from a technic, economic and environmental point of view. It would be also interesting to study these dielectric fluids with other heat sink configurations like the ones defined as pin-fin heat sinks, which are also widely used in Power Electronics.

9 References

(1) Guan, Dazhong, 2011. Thermal Analysis of Power Electronic Modules. Doctoral thesis, Technical Institute of Physics and Chemistry, Chinese Academy of Sciences, 2011.

(2) Solvay. Galden® PFPE Perfluoropolyether Fluorinated Fluids. [online] [Consulted on 21/03/2018] Available in: <https://www.solvay.com/en/markets-and-products/featured-products/Galden.html>.

(3) Solvay. Galden® HT PFPE. Heat Transfer Fluids. Specialty Polimers. [online] [Consulted on 21/03/2018] Available in: <https://www.solvay.com/en/binaries/Galden-PFPE-Heat-Transfer-Fluids_EN-220543.pdf>

(4) 3M. Heat transfer applications using 3M™ Novec™ Engineered Fluids Electronics Materials Solutions Division. [online] [Consulted on 21/03/2018] Available in: <https://multimedia.3m.com/mws/media/1091997O/3m-novec-engineered-fluids-for-heat-transfer-line-card.pdf>.

(5) 3M. 3M Novec™ 7500 Engineered Fluid. Electronic Markets Materials Division. [online] [Consulted on 21/03/2018] Available in <http://multimedia.3m.com/mws/media/65496O/3mtm-novectm-7500-engineered-fluid.pdf>

(6) 3M. Fluorinert™ Electronic Liquid FC-3283. 3M Specialty Materials. [online] [Consulted on 21/03/2018]. Available in: <http://multimedia.3m.com/mws/media/64886O/fluorinert-electronic-liquid-fc-3283.pdf?fn=prodinfo_FC3283.pdf>

(7) Prof. Dr.-Ing. März, Martin. Thermisches Management in der Leistungselektronik. Friedrich-Alexander-Universität Erlangen –Nürnberg. Fraunhofer Institute for Integrated Systems and Device Technology IISB. 2017

(8) Velo, Enrique. Disseny d'Equips Tèrmics. Convecció. E.T.S. Enginyeria Industrial de Barcelona. Departament de Màquines I Motors Tèrmics. UPC Barcelona Tech. 2017.

(9) Knight, R. W., Goodling, J. S., Hall, D. J. Optimal Thermal Design of Forced Convection Heat Sinks – Analytical. The American Society of Mechanical Engineers, 1991.

(10) ANSYS®, Inc. Products. Fluids. ANSYS Fluent. [online] [Consulted on 05/04/2018] Available in: <https://www.ansys.com/products/fluids/ansys-fluent>.

(11) Quora. What is the difference between pressure based solver and density based solver in Ansys Fluent? What is the criteria while selecting one of them? [on line] [Consulted on 06/04/2018] Available in: <https://www.quora.com/What-is-the-difference-between-pressure-based-solver-and-density-based-solver-in-Ansys-Fluent-What-is-the-criteria-while-selecting-one-of-them>

(12) Wasserman, Shawn. engineering.com. Simulation. Choosing the Right Turbulence Model for Your CFD Simulation. [online] [Consulted on 06/04/2018] [Available in: <https://www.engineering.com/DesignSoftware/DesignSoftwareArticles/ArticleID/13743/Choosing-the-Right-Turbulence-Model-for-Your-CFD-Simulation.aspx>

(13) Glycol Heat-Transfer Fluids Ethylene Glycol versus Propylene Glycol. [online] [Consulted on 16/04/2018] [Available in: <http://www.veoliawatertech.com/crownsolutions/ressources/documents/2/21823,Glycol.pdf>]

(14) Microelectronics Heat Transfer Laboratory. Fluid Properties Calculator. Ethylene glycol. [online] [Consulted on 16/04/2018] [Available in: <http://www.mhtl.uwaterloo.ca/old/onlinetools/airprop/airprop.html>]